Chapter 18

Process equipment in petroleum refining

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Introduction

This chapter deals with the items of equipment normally met with in the petroleum refining industry. Indeed, many of the items that will be described and discussed here are also common to many other process industries. Knowledge of these equipment items are essential for good refinery design, operation, and troubleshooting when necessary. The equipment described here falls into the following categories, and will be presented in the following parts:

Part 1 Vessels
Part 2 Pumps
Part 3 Compressors
Part 4 Heat Exchangers
Part 5 Fired Heaters

These sections will include a description of the various types, an in depth discussion, and design features. They will also provide an example of the data sheet usually forwarded to manufacturers for the items required. Invariably in refinery technical libraries these data sheets are included as part of the 'Mechanical Catalogues' and supported by narrative specifications which give details of metallurgy and fabrication codes etc. These catalogues are provided by the equipment supplier and are part of all the information dossier on each item. Included also are such items as installation details, start-up procedures, routine maintenance procedures and the like. In most refineries today the catalogues are kept on computer discs or microfilm.

18.1 Vessels

This section address the pressure vessels that are common to most refineries. These include:

- Columns and Towers
- Knock out Drums and Separators
- Accumulators and Surge vessels

Storage tanks have been dealt with in Chapter 13 of this Handbook.

Fractionators, trays, and packings

Trayed towers

Columns normally constitute the major cost in any chemical process configuration. Consequently it is required to exercise utmost care in handling this item of equipment. This extends to the actual design of the vessel or evaluating a design offered by others. Normally columns are used in a process for fractionation, extraction or absorption as unit operations. Columns contain internals which may be trays, or packing. Both types of columns will also contain suitable inlet dispersion nozzles, outlet nozzles, instrument nozzles, and access facilities (such as manholes or handholes). This item deals with the trayed towers.

Tray types

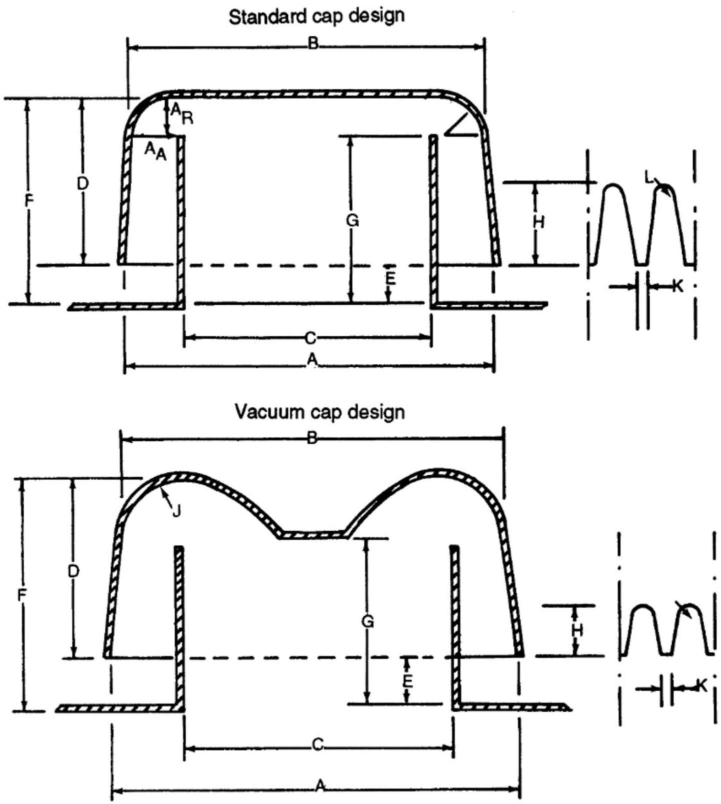
There are three types of trays in common use today. These are:

- Bubble cap
- Sieve
- Valve

Bubble cap trays
This type of tray was in wide use up until the mid to late 1950s. Their predominance was displaced by the cheaper sieve and valve trays. The bubble cap tray consists of a series of risers on the tray which are capped by a serrated metal dome. Figure 18.1 shows two types of caps. One is used in normal fractionation service while the other is designed for vacuum distillation service. Vapor rises up through the risers into the bubble cap. It is then forced down through the serrated edge or, in some cases, slots at the bottom of the cap. A liquid level is maintained on the tray to be above the slots or serrations of the cap. The vapor therefore is forced out in fine bubbles into this liquid phase thereby mixing with the liquid. Mass and heat transfer between vapor and liquid is enhanced by this mixing action to effect the fractionation mechanism.

Capacity. Moderately high with high efficiency.

Efficiency. Very efficient over a wide capacity range.



Dimension	6"	6" raised	4"	6" vacuum
A	6"	6"	4"	6"
B	$5^3/_4$"	$5^3/_4$"	$3^3/_4$"	$5^3/_4$"
C	4"	4"	$2^1/_2$"	4"
D	$2^3/_4$"	$2^3/_4$"	$2^1/_8$"	$2^3/_4$"
E	$^5/_8$"	$^3/_4$"	$^1/_2$"	$^3/_4$"
F	$3^3/_8$"	$3^1/_2$"	$2^5/_8$"	$3^1/_2$"
G	$2^3/_4$"	$2^1/_2$"	$2^1/_4$"	$2^1/_2$"
H	$1^3/_8$"	$1^3/_8$"	1"	$^3/_4$"
J	$^5/_8$"	$^5/_8$"	$^3/_8$"	1"
K	$^5/_{32}$"	$^5/_{32}$"	$^5/_{32}$"	$^5/_{32}$"
L	$^5/_{32}$"	$^5/_{32}$"	$^1/_8$"	$^5/_{32}$"
Number of slots	28	28	20	28
Total slot area/cap (ft^2)	0.105	0.105	0.047	0.056
Peripheral area under cap (ft^2)	0.082	0.098	0.044	0.098
Total effective slot area (ft^2)	0.187	0.203	0.091	0.154
Chimney area (ft^2)	0.087	0.087	0.034	0.087

Figure 18.1. Bubble cap design.

Entrainment. Much higher than perforated type trays due to the “Jet” action that accompanies the bubbling.

Flexibility. Has the highest flexibility both for vapor and liquid rates. Liquid heads are maintained by weirs.

Application. May be used for all services except for those conditions where coking or polymer formation occur. In this case Baffle or Disc & Donut trays should be used.

Note: Because of the relatively high liquid level required by this type of tray it incurs a higher pressure drop than most other types of trays. This is a critical factor in tray selection for vacuum units.

Tray Spacing. Usually 18″ to 24″. For Vacuum service this should be about 30″ to 36″.

Sieve trays

This is the simplest of the various types of trays. It consists of holes suitably arranged and punched out of a metal plate. The vapor from the tray below rises through the holes to mix with the liquid flowing across the tray. Fairly uniform mixing of the liquid and vapor occurs and allows for the heat/mass transfer of the fractionation mechanism to occur. The liquid flows across a weir at one end of the tray through a downcomer to the tray below.

Capacity. As high as or higher than bubble cap trays at design vapor/liquid rates. Performance drops off rapidly at rates below 60% of design.

Efficiency. High efficiency at design rates to about 120% of design. The efficiency falls off rapidly to around 50–60% of design rate. This is due to “weeping” which is the liquid leaking from the tray through the sieve holes.

Entrainment. Only about one-third of that for bubble cap trays.

Flexibility. Not suitable for trays operating at variable loads.

Application. In most mass transfer operations where high capacities in vapor and liquid rates are required. Handles suspended solids and other fouling media well.

Tray spacing. Requires less tray spacing than bubble cap. Usually spacing is rarely less than 15″ although some services can operate at 10″ and 12″. In vacuum service a spacing of 20″ to 30″ is acceptable.

Valve trays

These trays have downcomers to handle the liquid traffic and holes with floating caps that handle the vapor traffic. The holes may be round or rectangular and the caps over the holes are moveable within the limits of the length of the “legs” which fit into the

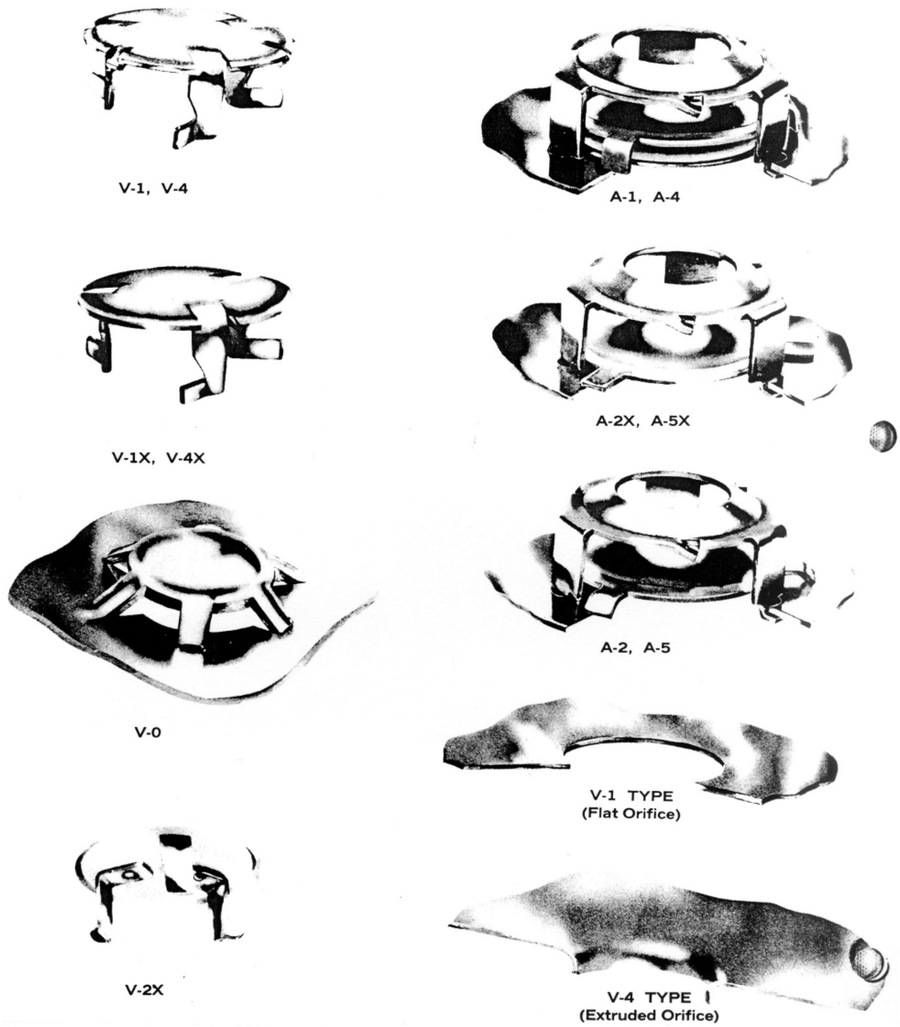


Figure 18.2. Valve unit types—Glitsch Ballast.

holes. Figures 18.2 and 18.3 show the type of valves and valve trays offered by Glitsch as their "Ballast" trays.

Valve trays are by far the most common type of tray used in the chemical industry today. The tray has good efficiency and a much better flexibility in terms of turn down than the sieve or bubble cap trays. Its only disadvantage over the sieve tray is that it is slightly more expensive and cannot handle excessive fouling as well as the sieve tray. The remainder of this item will now be dedicated to the sizing and analysis of the valve tray tower.

Description of Ballast® Units

The various types of Ballast units are shown on the page 881. A description of each unit follows:

V-0 A non-moving unit similar in appearance to the V-1 in a fully open position. It is used in services where only moderate flexibility is required and minimum cost is desired.

V-1 A general purpose standard size unit, used in all services. The legs are formed integrally with the valve for deck thicknesses up to ⅜″.

V-2 The V-2 unit is similar to the V-1 unit except the legs are welded-on in order to create a more leak-resistant unit. The welded legs permit fabrication of Ballast units for any deck thickness or size. Large size units are frequently used for replacement of bubble caps.

V-3 A general purpose unit similar to the V-2 unit except the leg is radial from the cap center.

V-4 This signifies a venturi-shaped orifice opening in the tray floor which is designed to reduce substantially the parasitic pressure drop at the entry and reversal areas. A standard Ballast unit is used in this opening normally, although a V-2 or V-3 unit can be used for special services. The maximum deck thickness permissible with this opening is 10 gage.

V-5 A combination of V-0 and V-1 units. It normally is used where moderate flexibility is required and a low cost is essential.

A-1 The original Ballast tray with a lightweight orifice cover which can close completely. It has a separate Ballast plate to give the two-stage effect and a cage or travel stop to hold the Ballast plate and orifice cover in proper relationship.

A-2 The same as A-1, except the orifice cover is omitted.

A-4 An A-1 unit combined with a venturi-shaped orifice opening in order to reduce the pressure drop.

A-5 An A-2 unit combined with a venturi-shaped opening.

The diameter of the standard size of the V-series of Ballast units is 1⅞″. The V-2 and V-3 units are available in sizes up to 6″.

Photographs of several Ballast trays are shown on page 8 and 9.

Figure 18.2. (Cont.)

Trayed tower sizing

The height of a trayed tower is determined by the number of trays it contains, the liquid surge level at the bottom of the tower, and the tray spacing. The number of trays is a function of the thermodynamic mechanism for the fractionation or absorption duty required to be performed. This is described in Chapters 3 and 4 of this Handbook.

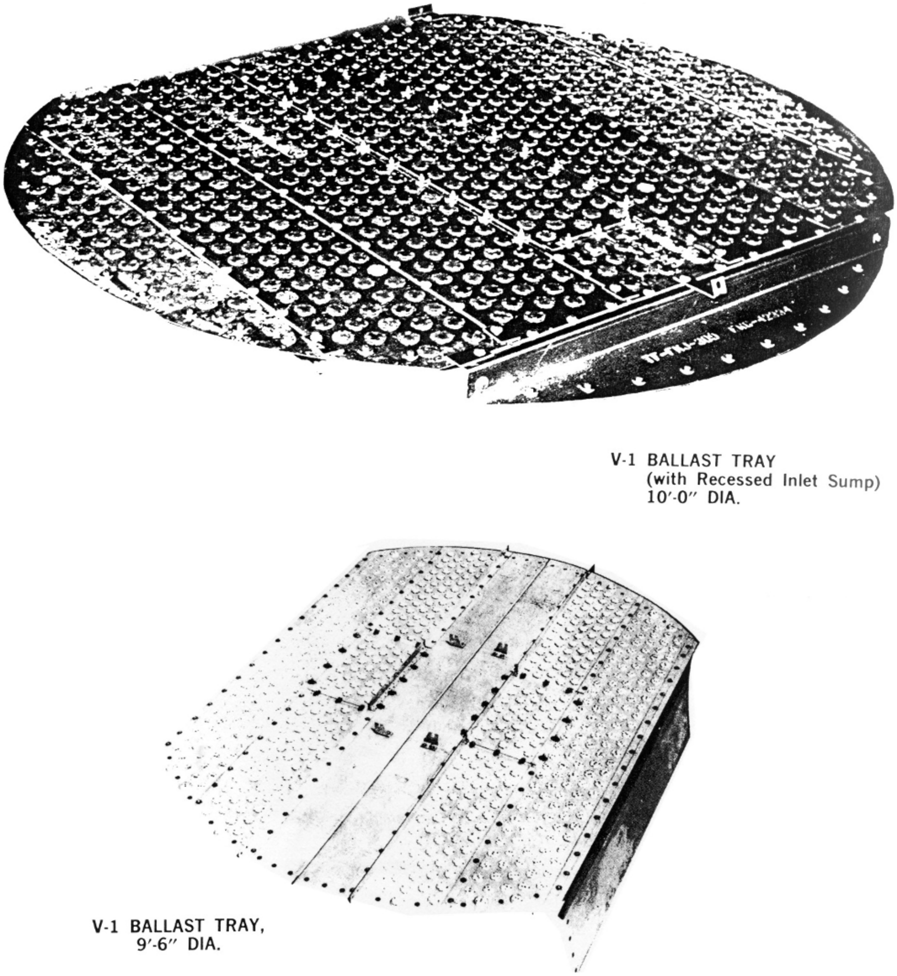


Figure 18.3. Valve trays—Glitsch Ballast.

The diameter of the tower is based on allowable vapor and liquid flow in the tower and the type of tray. This section now deals with determining the tower diameter using valve trays.

The "quickie" method

This method is good enough for a reasonable estimate of a tower diameter which can be used for a budget type cost estimate or initial plant layout studies. The steps used for this calculation are as follows:

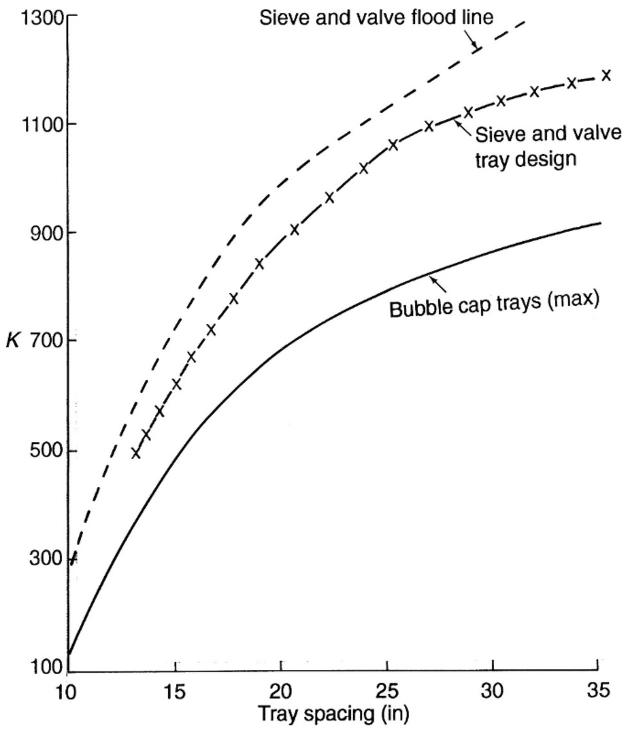


Figure 18.4. Tray spacing versus 'K' factor.

Step 1. Establish the liquid and vapor flows for the critical trays in the section of the tower that will give the maximum values. These are obtained by heat balances (see Chapters 3 and 4 of this Handbook). The critical trays are:

The top tray.
Any side stream draw off tray.
Any intermediate reflux draw off tray.
The bottom tray.

Step 2. Calculate the actual cuft/hr at tray conditions of the vapor. Then using the total mass per hour of the vapor calculate its vapor density in lbs/cuft.

Step 3. From the heat balance determine the density of the liquid on the tray at tray temperature in lbs/cuft.

Step 4. Select a tray spacing. Start with a 24″ space. Read from Figure 18.4 a value for 'K' on the flood line. Using the equation:

$$G_f = K\sqrt{((\rho_v \times (\rho_l - \rho_v))}$$

where

G_f = mass vapor velocity in lbs/hr sqft at flood,
K = the constant read from the flood curve in Figure 18.4,
ρ_v = density of vapor at tray conditions in lbs/cuft,
ρ_L = density of liquid at tray conditions in lbs/cuft.

Step 5. Multiply G_f by 0.82 to give mass velocity at 82% of flood which is the normal recommended design figure. Divide the actual vapor rate in lbs/hr by the vapor mass velocity to give the area of the tray. Calculate tray diameter from this area.

An example calculation now follows.

Example calculation
Calculate the diameter of the tower to handle the liquid and vapor loads as follows:

Vapor to tray	Liquid from tray
lbs/hr = 47,700	GPH @ 60 = 119.7
moles/hr = 929.7	Hot GPH = 153.0
ACFS = 7.83	Hot CFS = 0.339
lbs/cuft ρ_V = 1.69	lbs/hr = 33,273
Temp °F = 167	lbs/cuft ρ_L = 27.3
Pressure PSIA = 220	Temp °F = 162

Tray spacing is set at 24″ and the trays are valve type.

From Figure 18.4 'K' = 1,110 at flood.

$$\rho_v = 1.69 \text{ lbs/cuft at tray conditions.}$$

To calculate ρ of vapor at tray conditions use:

$$\rho = \frac{\text{wt/hr} \times 520 \times \text{pressure (psia)}}{378 \times 14.7 \times \text{moles/hr} \times \text{Temp } ^\circ\text{R}}$$

where

Press is tray pressure.
Temp °R is tray temperature in °F + 460.

$$\begin{aligned}
\rho_L &= 27.3 \text{ lbs/cuft.} \\
\text{Then } G_f &= 1{,}110\sqrt{1.69 \times (27.3 - 1.69)} \\
&= 7{,}302 \text{ lbs/hr.sqft} \\
\text{Area of tray @ 82\% of flood} &= \frac{47{,}700}{0.82 \times 7{,}302} \text{ sqft} \\
&= 7.97 \text{ sqft} \\
&= 3.18 \text{ ft. say } 39''
\end{aligned}$$

The tray dimensions and configuration for design purposes are subject to a much more rigorous examination. This is normally undertaken by the tray manufacturer from data supplied by the purchaser's process engineer. However this process engineer needs to be able to check the manufacturer's offer before committing to purchase. The following calculation procedure offers a rigorous method for this purpose which establishes tray size and geometry. This calculation is based on a method developed by Glitsch Inc., a major manufacturer of valve and other types of trays and packing.

The rigorous method

A rigorous method used in the design of valve trays is described by the following calculation steps:

Step 1. Establish the liquid and vapor flows as described earlier for the "quickie" method.

Step 2. Calculate the down comer Design Velocity V_{dc} using the following equations (or by Figure 18.5).

(a) $V_{dc} = 250 \times \text{system factor}$
(b) $V_{dc} = 41 \times \sqrt{(\rho_L - \rho_v)} \times \text{system factor.}$
(c) $V_{dc} = 7.5 \times \sqrt{TS} \times \sqrt{(\rho_L - \rho_v)} \times \text{system factor}$

where TS = Tray spacing, in inches
Use the lowest value for the design velocity in gpm/sqft.
Down comer System Factors are given in Table 18.1.

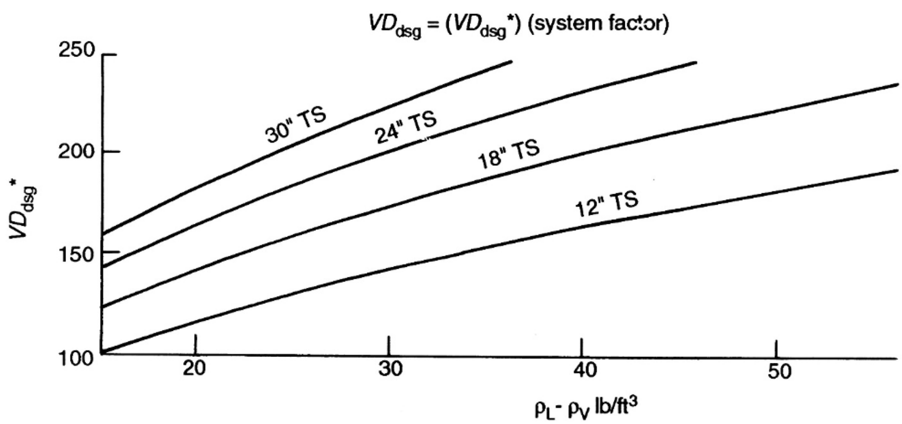


Figure 18.5. Down-comer design velocity.

Table 18.1. Down comer system factors

Service	System factor
Non foaming, regular system	1.0
Fluorine systems	0.9
Moderate foaming (amine units)	0.85
Heavy foaming (glycol, Amine)	0.73
Severe foaming (MEK units)	0.6
Foam stable systems (caustic regen)	0.3

Step 3. Calculate the Vapor Capacity Factor CAF using Figure 18.6.

$$\text{CAF} = \text{CAFo} \times \text{system factor.}$$

System factors used for this equation are given in Table 18.2.

Step 4. Calculate the vapor load using the equation:

$$V_1 = \text{CFS}\sqrt{\rho_v/(\rho_L - \rho_v)}$$

where CFS = actual vapor flow in cuft/sec.

Step 5. Establish tower diameter using Figure 18.7. Tray spacing is usually 18″, 24″, or 30″ for normal towers operating at above atmospheric pressures. Large vacuum towers may have tray spacing 30 to 36″. Note this diameter may be increased if other criteria of tray design are not met.

Step 6. Calculate the approximate Flow Path Length (FPL) based on tower diameter from Step 5 using the equation:

$$\text{FPL} = 9 \times \text{DT/NP}$$

where

FPL = Flow Path Length in ins.
DT = Tower Diameter from step 5 in ft
NP = Number of passes. For small towers with moderate liquid flows this will be 1. For larger towers this will depend on liquid velocities in the down comer. The highest number of passes is usually 4.

Step 7. Calculate the *minimum* active area (AA_m) using the expression:

$$AA_m = \frac{V_1 + (L \times \text{FPL}/13{,}000)}{\text{CAF} \times \text{FF}}$$

where

AA_m = Minimum active area in sqft.
V_1 = Vapor load in CFS.
L = Liquid flow in actual gpm.

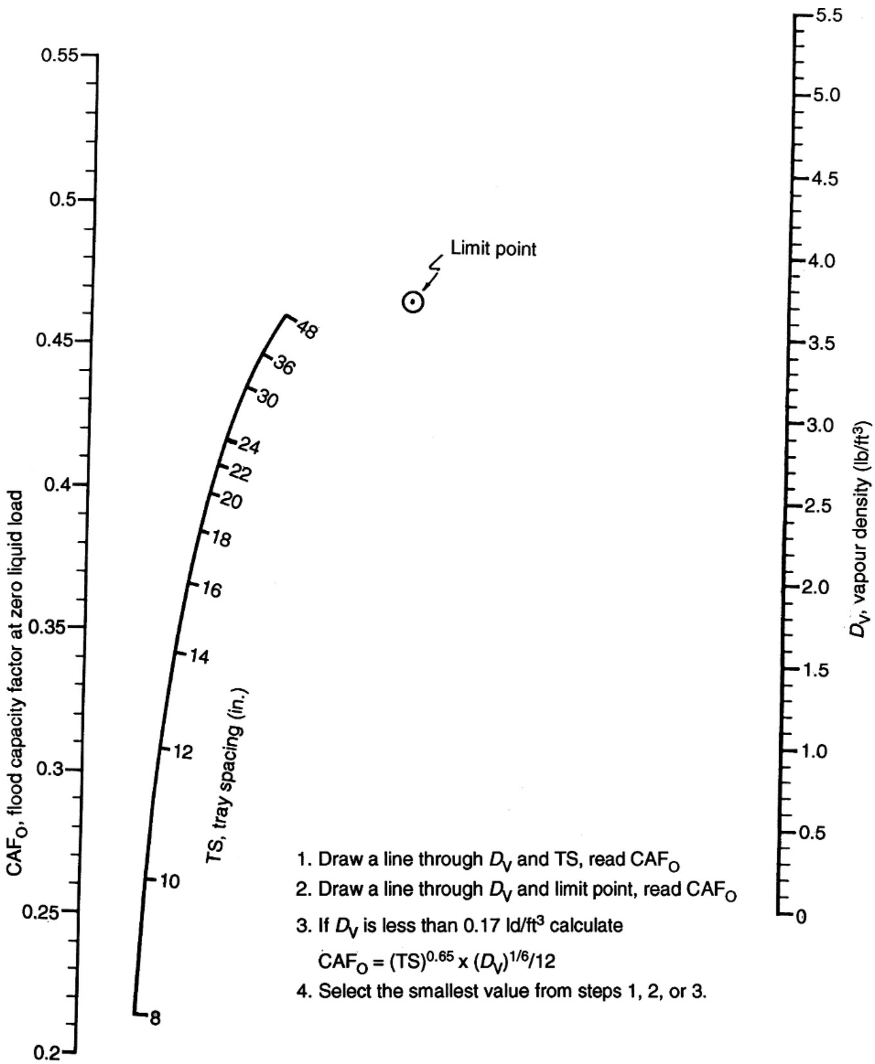


Figure 18.6. Flood capacities of valve trays.

FPL = Flow path length in ins.
CAF = Capacity Factor from Step 3.
FF = Flood Factor usually 80–82%.

Step 8. Calculate *minimum* down comer area (AD_m) using the equation

$$AD_m = \frac{L}{V_{dc} \times 0.8}$$

Table 18.2. Vapor system factors

Service	System factor
Non-foaming, regular	1.0
Fluorine systems	0.9
Moderate foaming	0.85
Heavy foaming	0.73
Severe foaming	0.6
Foam stable system	0.3–0.6

where

AD_m = minimum down comer area in sqft
L = Actual liquid flow in gpm
V_{dc} = Design down comer velocity from step 2.

Note: The down comer liquid velocity using the calculated minimum down comer area should be around 0.3 to 0.4 ft/sec.

Step 9. Calculate the *minimum* tower cross-sectional area using the following equations:

$$AT_m = AA_m + 2AD_m$$

or

$$AT_m = \frac{V_l}{0.78 \times CAF \times 0.8}$$

where

AT_m = minimum tower cross-sectional area in sqft.

Step 10. Calculate actual down comer area using the following equation:

$$AD_c = \frac{AT \times AD_m}{AT_m}$$

where

AD_c = Actual down comer area in sqft.
AT = Tower area in sqft from the diameter calculated in step 5.

Step 11. Determine down comer width (H_1) (from Table 18.A.1 in the Appendix of this Chapter) for side down comers. For multipass trays use the following equation with

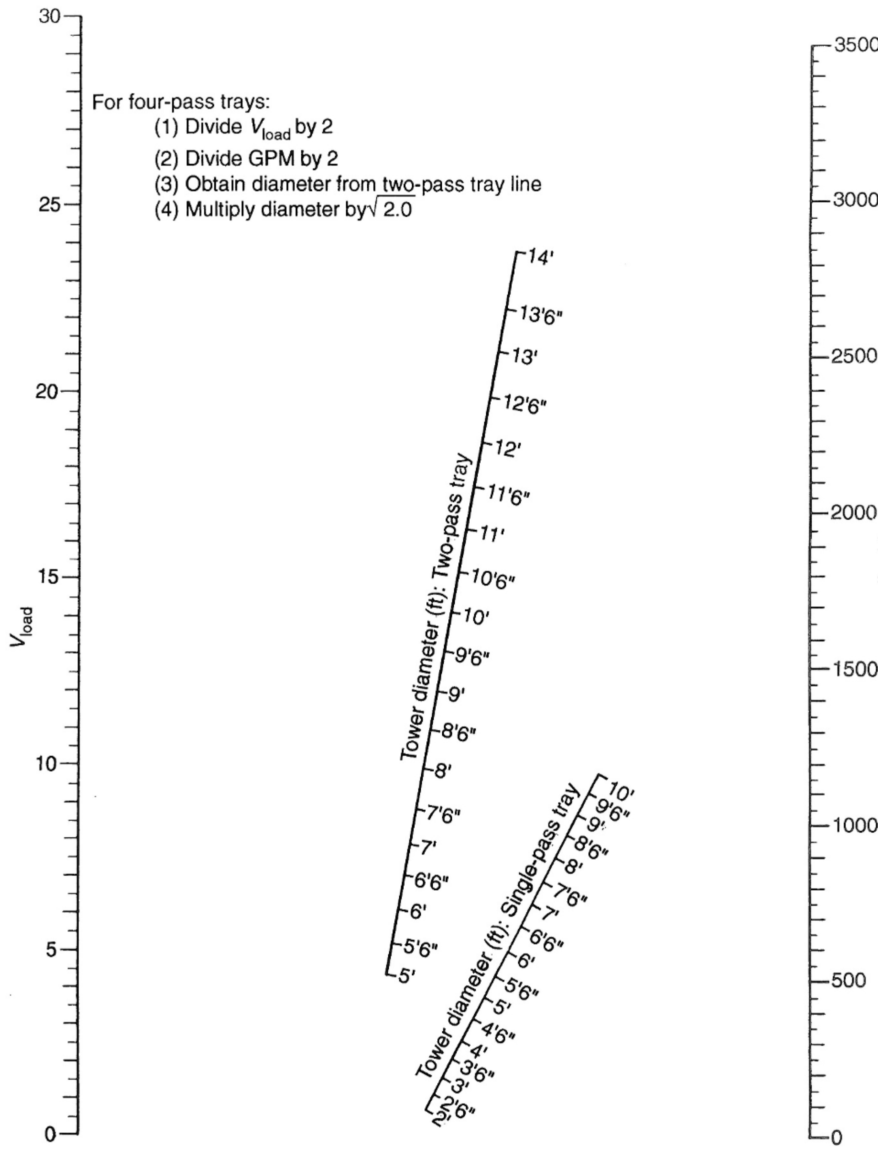


Figure 18.7. Tray diameter versus vapor loads.

Table 18.3. Allocation of down comer area and width factors

	←	Fraction of Total Down ComerArea →		
No of passes	AD1	AD2	AD3	AD4
1	0.5% each	1.0	–	–
2	0.34% each	–	0.66	–
3	0.25% each	0.5	0.5 each	–
4	0.2	–	0.4	0.4
		Width Factors (WF)		
Passes	H4	H5	H7	
1	12.0	–	–	
2	–	8.63	–	
3	6.0	6.78 each pass	–	
4	–	5.66	5.5	

the width factors given in Table 18.3:

$$H_i = WF \times \frac{AD}{DT}$$

where

H_i = width of individual down comers in ins
WF = width factor from Table 18.3
AD = total down comer area in sqft
DT = actual tower diameter in ft

See Figure 18.8 for allocation of down comers in multi pass trays.

Step 12. Calculate the actual FPL from the equation:

$$FPL = \frac{12 \times DT - (2H_1 + H_3 + 2H_5 + 2H_7)}{NP}$$

where

H_{1-7} are individual down comer widths in ins (see Figure 18.8)
NP = number of passes.

Step 13. Calculate actual active area (AA) using values for H calculated in step 12 and Table 18.A.1 in the Appendix to establish inlet areas of multi pass tray down comers.

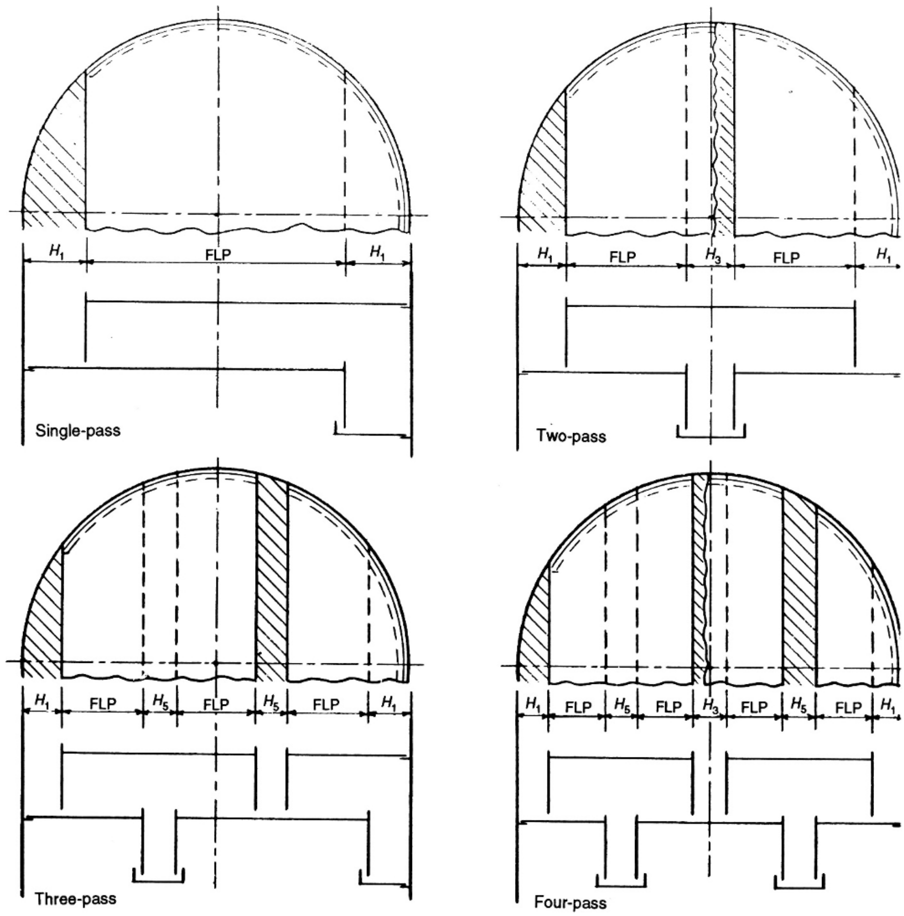


Figure 18.8. Types of tray.

$$AA = AT - (2AD_1 + AD_3 + 2AD_5 + 2AD_7)$$

where

AA = Actual active area in sqft.
AT = actual tower area in sqft.
AD_{1-7} are individual down comer areas in sqft for multi pass trays corresponding to H_{1-7}.

Step 14. From the data now developed calculate the actual percent of flood or flood factor (FF).

The following expression is used for this:

$$\%\,\text{Flood} = \frac{V_l + (L \times FPL/13{,}000) \times 100}{AA \times CAF}$$

Step 15. Calculate vapor hole velocity V_h assume 12–14 units (holes) per sqft of AA. Then

$$V_h = \frac{CFS \times 78.5}{NU}$$

where

V_h = Hole velocity in ft/sec.
CFS = Actual cuft/sec of the vapor.
NU = Total number of units.

Step 16. Calculate dry tray pressure drops from:

$$\text{Valves partly open:}\quad \Delta P_D = 1.35\, t_m \rho_m/\rho_l + K_1(V_h)(\rho_v/\rho_l)$$

where

ΔP_D = dry tray valve pressure drop in ins liquid
t_m = valve thickness in ins (see Table 18.4)
ρ_m = valve metal density in lbs/cuft (see Table 18.4)
K_1 = pressure drop coefficient (see Table 18.4)

Valves fully open: $\Delta P_D = K_2(V_h)^2 \rho_v/\rho_l$

where K_2 = pressure drop coefficient (see Table 18.4).

Table 18.4. Pressure drop coefficients (Glitch Ballast Type Trays)

		← K_2 for deck thickness (ins) →			
Type of Unit	K1	0.074	0/104	0.134	0.25
V1	0.2	1.05	0.92	0.82	0.58
V4	0.1	0.68	0.68	0.68	——

← Thickness →		← Densities of valve material →	
Gauge	tm (ins)	Metal	Density (lbs/cuft)
20	0.37	CS	400
18	0.5	SS	500
16	0.6	Ni	553
14	0.074	Monel	550
12	0.104	Titanium	283
10	0.134	Hasteloy	560
		Aluminum	168
		Copper	560
		Lead	708

Step 17. Calculate total tray pressure drop.

$$\Delta P = \Delta P_D + 0.4(L/L_{wi})^{0.87} + 0.4H_w$$

where

ΔP = total tray pressure drop in ins of liquid.
L_{wi} = weir length in ins (from Chapter 3 Appendix Figure A4.0).
H_w = weir height in ins (usually 1–2).

Step 18. Calculate height of liquid in down comer.
First calculate the head loss under the down comer H_{UD}, where $H_{UD} = 0.65(V_{UD})^2$. V_{UD} is calculated from the liquid velocity in CFS or gpm/450 divided by the area under the down comer. Use weir length times weir height for the area. This velocity should be around 0.3–0.6 ft/sec for most normal towers.

Then:

$$H_{dc} = H_w + 0.4(L/L_{wi})^{.67} + (\Delta P + H_{UD})(\rho_l/(\rho_l - \rho_v)).$$

where

H_{dc} = height of liquid in down comer in ins
For normal design this should not exceed 50% of tray spacing.

An example calculation now follows:

Example calculation
In this example the same Liquid and Vapor flows and data will be used as used in the 'Quickie Calculation'. The objective of this calculation will be to determine the tower diameter, tray pressure drop and configuration and the percent flood for the design flow rates given.

Calculating the down comer design velocity V_{dc}
System factor in this case is 1.

$$V_{dc} = 250 \times 1.0 = 250$$

or $$V_{dc} = 41 \times (\rho_1 - \rho_v) \times 1.0 = 41 \times 5.06 \times 1.0 = 207$$

or $$V_{dc} = 7.5 \times \text{TS} \times (\rho_1 - \rho_v) \times 1.0 = 7.5 \times 4.9 \times 5.06 = 186\,(\text{TS} = \text{tray spacing} = 24'')$$

or $V_{dc} = 188$ from Figure 18.5, use $V_{dc} = 186$ gpm/sqft.

Vapor capacity factor CAF.

System factor in this case is 1.
From Figure 18.6 CAF = 0.43.

Actual vapor load V_l.

$$\begin{aligned} V_{\mathrm{l}} &= \mathrm{CFS}\sqrt{\rho_{\mathrm{v}}/(\rho_{\mathrm{l}} - \rho_{\mathrm{v}})} \\ &= 7.83\sqrt{1.69/(27.3 - 1.69)} \\ &= 2.01. \end{aligned}$$

Approximate tower diameter AT.

$$\begin{aligned} \text{Using Figure 9.7 } V_{\mathrm{l}} &= 2.01 \\ \mathrm{TS} &= 24'' \\ L &= 153\,\mathrm{gpm} \\ \text{Tower id} &= 3.25\,\mathrm{ft} = 39'' \\ \text{Area} &= 8.30\,\mathrm{sqft.} \end{aligned}$$

Calculate approx flow path length FPL.

$$\mathrm{FPL} = \frac{9 \times 39}{\text{No of passes}} \times 12 = 29.25\,\mathrm{ins.}$$

Calculate minimum active area AA_m.

$$\begin{aligned} \mathrm{AA_m} &= \frac{V_{\mathrm{l}} + (L \times \mathrm{FPL}/13{,}000)}{\mathrm{CAF} \times \mathrm{FF}} \\ &= \frac{2.01 + (153 \times 29.25/13{,}000)}{0.43 \times 0.8} \qquad \text{(using 80\% flood)} \\ &= 3.01\,\mathrm{sqft.} \end{aligned}$$

Calculating minimum down comer area AD_m.

$$\begin{aligned} \mathrm{AD_m} &= \frac{\mathrm{L}}{V_{\mathrm{dc}} \times 0.8} \\ &= \frac{153}{186 \times 0.8} = 1.028\,\mathrm{sqft.} \end{aligned}$$

Calculating the minimum tower cross-sectional area.

$$\begin{aligned} \text{Either } \mathrm{AA_m} + 2\mathrm{AD_m} &= 3.01 + 2.056 \\ &= 5.066\,\mathrm{sqft.} \\ \text{or} \qquad \frac{V_{\mathrm{l}}}{0.78 \times \mathrm{CAF} \times 0.8} &= 7.49\,\mathrm{sqft} \end{aligned}$$

Use the larger which is 7.49 sqft.

Min diameter therefore is 3.09 ft say 37″.

Calculating actual downcomer area AD_c.

$$AD_c = \frac{AT \times AD_m}{AT_m} = \frac{8.3 \times 1.028}{7.49}$$
$$= 1.14\,\text{sqft.}$$

Down comer width $H = AD/AT = 1.14/8.30 = 0.137.$

From Chapter 3 Appendix Figure A4.0 H/D = 0.197 then

$$H = 0.197 \times 3.25 = 0.633\ \text{ft} = 76''.$$

Recalculating flow path length FPL.

$$\begin{aligned} \text{FPL} &= 12 \times D_t - (2H) \\ &= 12 \times 3.25 - (2 \times 7.6) \\ &= 23.8\ \text{ins.} \end{aligned}$$

Recalculating active area based on actual down comer area.

$$\begin{aligned} \text{AA} &= \text{AT} - (2\text{AD}) \\ &= 8.3 - 2.28 \\ &= 6.02\ \text{sqft. Which is greater than min allowed.} \end{aligned}$$

Checking percent of flood.

$$\begin{aligned} \%\ \text{flood} &= \frac{V_1 + (L \times \text{FPL}/13{,}000)}{\text{AA} \times \text{CAF}} \times 100 \\ &= \frac{2.01 + (153 \times 23.8/13{,}000)}{6.02 \times 0.43} \times 100 \\ &= 88.0\% \quad \text{Which is a little high for design but is acceptable.} \end{aligned}$$

Check down comer velocity.

CFS of liquid = 0.34 cfs
Area of down comer = 1.14 sqft

$$\text{Velocity of liquid in down comer} = \frac{0.34}{1.14} = 0.3\ \text{ft/sec.}$$

Calculating pressure drops and down comer liquid height.

- *Dry tray pressure drop.*
  Partially open valves.

$$\Delta P_D = 1.35\, t_m \rho_m/\rho_l + K_1 (V_h)^2 (\rho_v/\rho_l)$$

$$V_h = \frac{7.83 \times 78.5}{72} \qquad \text{(assumes 12 units per sqft of AA).}$$

$$= 8.5 \text{ ft/sec}$$

$$\Delta P_D = 1.35 \times 0.74'' \times (490/27.3) + (0.2 \times 72.25 \times 0.062)$$

$$= 2.69'' \text{ liquid.}$$

  Fully Open valves.

$$\Delta P_D = \mathrm{K_2}\,(\mathrm{V_h})^2 \cdot (\rho_v/\rho_l)$$

$$= 0.92 \times 72.25 \times 0.062$$

$$= 4.12'' \text{ liquid. This will be used.}$$

- *Total tray pressure drop.*

$$\Delta P = \Delta P_D + 0.4\,(L/L_{wi})^{0.67} + (0.4 \times H_w)$$

H_w (weir height) is fixed at 2″.
L_{wi} (down comer length) is calculated from Appendix A, Chapter 3 as 30.9″.

$$\Delta P = 4.12 + 0.4\,(153/30.9)^{0.67} + 0.8$$

$$= 6.09'' \text{ of liquid.}$$

Height of liquid in down comer.

$$H_{dc} = H_w + 0.4\,(L/L_{wi})^{0.67} + (\Delta P + H_{UD})(\rho_l/\rho_l - \rho_v)$$

$$H_{dc} = 2 + 0.4 \times 1.16 + (6.09 + 0.405)(27.3/25.61)$$

$$= 10.08'' \text{ liquid. This is 42.0\% of tray spacing which is acceptable.}$$

(H_{UD} was calculated using a down comer outlet area of $L_{wi} \times 2''$ giving a velocity of 0.339 CFS divided by 0.429 sqft which is 0.79 ft/sec. H_{UD} is then $0.65\,(0.79)^2 = 0.405$).

Calculating the actual number of valves for tray layout. With truss lines parallel to liquid flow.

$$\text{Rows} = \left[\frac{\text{FPL} - 8.5}{0.5 \times \text{Base}} + 1\right] \text{NP}$$

where

Base = spacing of units usually 3.0″, 3.5″, 4.0″, 4.5″, or 6.0″.

$$\text{Units/row} = \frac{\text{WFP}}{5.75 \times \text{NP}} - (0.8 \times \text{number of Beams}) + 1$$

With truss lines perpendicular to liquid flow.

$$\text{Rows} = \left[\frac{\text{FPL} - (1.75 \times \text{N}_\text{o}\ \text{trusses} - 6.0)}{2.5}\right]\text{NP}$$

$$\text{Units/row} = \frac{\text{WFP}}{\text{Base} \times \text{NP}} - (2 \times \text{N}_\text{o}\ \text{Major Beams}) + 1$$

where

WFP = Width of flow path in ins.
= AA × 144/FPL.

Using a base pitch of 3.5″ the number of rows on the trays with trusses parallel to flow were calculated to be 9.7. Units per row and were then calculated to be 8.73. This gives total number of valves over the active area as 84.7. Thus number of valves per sqft of AA is 14. The assumption of 12 in the calculation (item 14) gives a more stringent design therefore the assumption is acceptable.

Calculation Summary

Tower diameter	= 3.25 ft or 39″
Down comer Area (ea)	= 1.14 sqft (single pass)
Active area	= 6.02 sqft.
Percent of flood	= 88
Tray spacing	= 24″
Down comer backup	= 42.0% of tray spacing.
Number of valves	= 85
Number of rows	= 10
Valve pitch	= 3.5″

Packed towers and packed tower sizing

Although trayed towers are generally the first choice for fractionation and absorption applications, there a number of instances where packed towers are preferable. For example on small diameter towers (below 3ft diameter) packed towers are generally cheaper and more practical for maintenance, fabrication, and installation. At the other end of the spectrum packing in the form of grids and large stacked packed beds have superceded trays in vacuum distillation towers whose diameter range up to 30 ft in some cases. This is because packing offers a much lower pressure drop than trays.

The packing in the tower itself may be stacked in beds on a random basis or in a defined structured basis. For towers up to 10–15 ft the packing is usually dumped or random packed. Above this tower size and depending on its application the packing

may be installed on a defined stacked or structured manner. For practical reasons and to avoid crushing the packing at the bottom of the bed the packing is installed in beds. As a rule of thumb packed beds should be around 15 ft in height. About 20 ft should be a maximum for most packed sections.

Properties of good packing are as follows:

- Should have high surface area per unit volume
- The shape of the packing should be such as to give a high percentage of area in active contact with the liquid and the gas or in the two liquid phases in the case of extractors
- The packing should have favorable liquid distribution qualities
- Should have low weight but high unit strength
- Should have low pressure drop, but high coefficients of mass transfer

Some data on the various common packing available commercially are given in Tables 18.5–18.7. Figure 18.9 shows a sectional layout of a typical packed tower. Note this tower has bed supports designed for gas distribution and includes intermediate weir liquid distributors between some of the beds.

Other salient points concerning packed towers are as follows:

1.0 Reflux ratios, flow quantities, and number of theoretical trays or transfer units are calculated in the same manner as for trayed columns.
2.0 Internal liquid distributors are required in packed towers to ensure good distribution of the liquid over the beds throughout the tower.
3.0 The packed beds are supported by grids. These are specially designed to ensure good flows of the liquid and the gas phases.
4.0 Every care must be taken in the design of the packed tower that the packing is always properly "wetted" by the liquid phase. Packing manufacturers usually quote a minimum wetting rate for their packing. This is usually around 2.0–2.5 gpm of liquid per sqft of tower cross section. Most companies prefer this minimum to be around 3.0–3.5 gpm/sqft (Tables 18.5 and 18.6).

Sizing a packed tower

The height of the tower is determined by the methods used to calculate the number of theoretical trays required to perform a specific separation. These have been discussed earlier in Chapter 1. A figure equivalent to the height of a theoretical tray is then calculated to determine the height of packing required. This is used as the basis to determine the overall height of the tower by adding in the space required for distributors, support trays and the like.

Table 18.5. Physical properties of some common packing

Packing type	Size (in.)	Wall thickness (in.)	OD and length	Approx. no./per ft^3	Approx. wt/per ft^3	Approx. surface area (ft^2 ft^3)	% void volume
Raschig rings (ceramic)	1/4	1/32	1/4	88 000	46	240	73
	5/16	1/16	5/16	40 000	56	145	64
	3/8	1/16	3/8	24 000	52	155	68
	1/2	1/32	1/3	10 600	54	111	63
	1/2	1/16	1/2	10 600	48	114	74
	5/8	3/32	3/8	5 600	48	100	68
	3/4	3/32	3/4	3140	44	80	73
	1	1/3	1	1350	40	58	73
	1 1/4	3/16	1 1/2	680	43	45	74
	1 1/2	1/4	1 2/3	375	46	35	68
	1 1/2	3/16	1 1/2	385	42	38	71
	2	1/4	2	162	38	28	74
	2	3/16	2	164	35	29	78
	3	3/8	3	48	40	19	74
Raschig rings (metal) (1)	1/4	1/32	1/4	88 000	150	236	69
	5/16	1/32	5/6	45 000	120	190	75
	5/16	1/16	5/16	43 000	198	176	60
	1/2	1/32	4/2	11 800	77	128	84
	1/2	1/16	1/2	11 000	132	118	73
	19/32	1/32	19/32	7 300	66	112	86
	19/32	1/32	19/32	7 000	120	106	75
	3/4	1/32	3/4	3400	55	84	88
	3/4	1/16	3/4	3190	100	72	78
	1	1/32	1	1440	40	63	92
	1	1/16	1	1345	73	57	85
	1 1/4	1/16	1 1/2	725	62	49	87
	1 1/2	1/16	1 1/2	420	50	41	90
	2	1/16	2	180	38	31	92
	3	1/16	3	53	25	20	95
Raschig rings (carbon)	1/4	1/16	1/4	85 000	46	212	55
	1/2	1/16	1/2	10 600	27	114	74
	3/4	1/8	3/4	3140	34	75	67
	1	1/3	1	1325	27	57	74
	1 1/4	1/16	1 1/4	678	31	45	69
	1 1/2	1/2	1 1/2	392	34	37	67
	2	1/5	2	166	27	28	74
	3	5/16	3	49	33	19	78
Berl saddles (ceramic)	1/4	–	–	113 000	56	274	60
	1/2	–	–	16 200	54	142	63
	3/4	–	–	5 000	48	82	66
	1	–	–	2200	45	76	69
	1 1/2	–	–	580	38	44	75
	2	–	–	250	40	32	72
Intalox saddles (ceramic)	1/4	–	–	117 500	54	300	75
	1/2	–	–	20 700	47	190	78
	3/4	–	–	6500	44	102	77
	1	–	–	2385	42	78	77
	1 1/2	–	–	709	37	60	81
	2	–	–	265	38	36	79

Table 18.6. Coefficients for use in the HETP equation

Packing type	Packing size (ins)	K_1	K_2	K_3
Raschig rings	0.375	2.10	− 0.37	1.24
	0.500	0.853	− 0.24	1.24
	1.000	0.57	− 0.10	1.24
	2.000	0.52	0.00	1.24
Saddles	0.500	5.62	− 0.45	1.11
	1.000	0.76	− 0.14	1.11
	2.000	0.56	− 0.02	1.11

The diameter of the tower is calculated using a method which allows for good mass and heat transfer while minimizing entrainment. The same principle of tower flooding is applicable to packed towers as for trayed towers. A calculation procedure for determining a packed tower diameter and the height of packed beds now follows:

Step 1. From examination of the flows of vapor and liquid in the tower determine the critical section of the tower where the loads are greatest. Usually this is at the bottom of an absorption unit and either the top or bottom of a fractionator.

Step 2. Determine the conditions of temperature and pressure at the critical tower section. This is usually accomplished by bubble and dew point calculations as described in Chapter 1. That is bubble point of the bottoms liquid (either in a fractionator or an absorber) determines the bottom of the tower conditions and dew point calculation of the overhead vapor determines the tower top conditions.

Step 3. Establish the liquid and vapor stream compositions at the critical tray conditions. See Chapter 3 for determining vapor/liquid streams in absorption and fractionation towers. Calculate the properties of these streams such as densities, mass/unit time, moles/unit time, viscosity, etc at the conditions of the critical tower section. Next select a packing type and size. Use Table 18.7 for this.

Table 18.7. Recommended packing sizes

Packing type	◀——Tower diameter ft——▶				
	1.0	2.0	3.0	4.0	+ 5.0
Raschig rings	0.5	0.75	1.00	1.5	3.00
Berl saddles	0.75	1.50	2.00	2.00	3.00
Interlox saddles	0.75	1.50	2.00	2.00	3.00
Pall rings	1.00	1.50	2.00	2.00	3.00

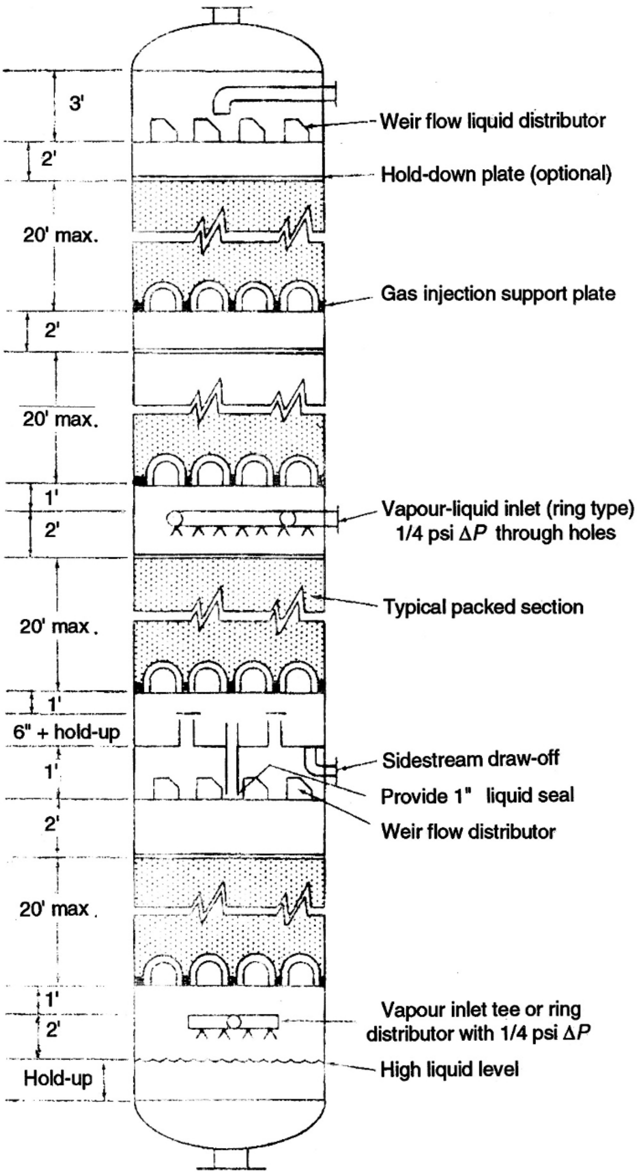


Figure 18.9. A typical packed tower.

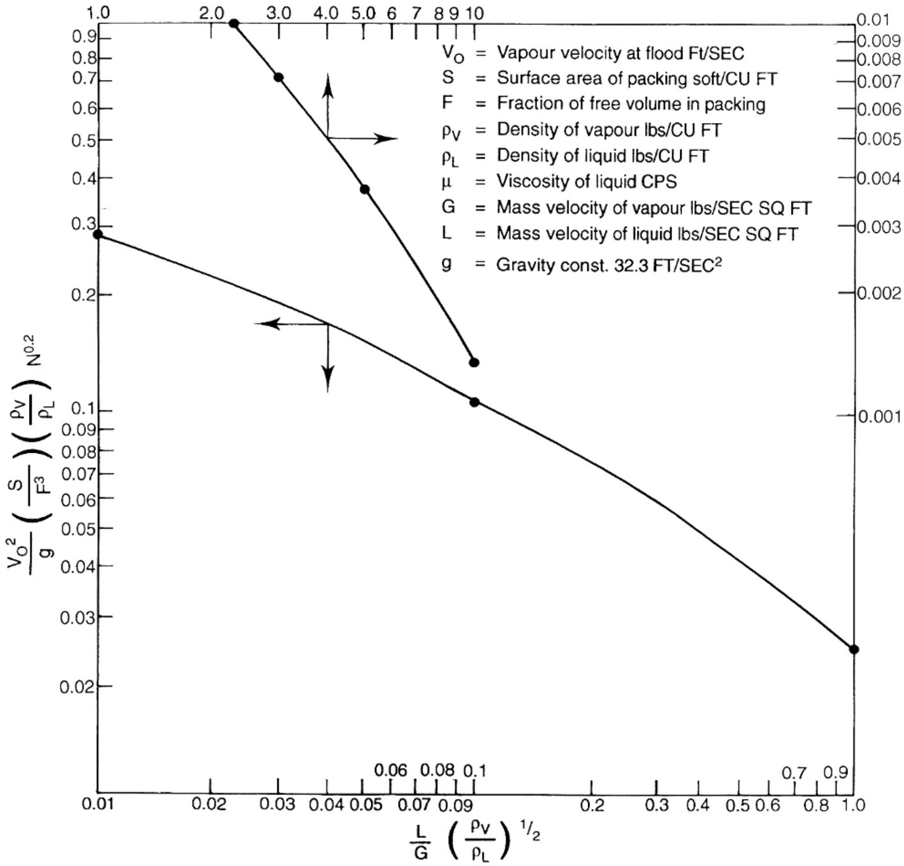


Figure 18.10. Packed tower flooding criteria.

Step 4. Commence the tower sizing by calculating the diameter. First calculate a value for

$$(L/G)\sqrt{(\rho_v/\rho_1)}$$

where

L = mass liquid load in lbs/sec sqft
G = mass vapor load in lbs/sec sqft
ρ_v = density of vapor in lbs/cuft @ tower conditions
ρ_1 = density of liquid in lbs/cuft @ tower conditions

Then using Figure 18.10 read off a value for the equation:

$$\left[\frac{L\rho_v}{G\rho_l}\right]^{1/2} = \frac{V^2 \cdot S \cdot \rho_v \cdot \mu^{0.2}}{g\,(F)^3 \rho_1}$$

where

V = the vapor velocity at flood in ft/sec
g = 32.2 ft/sec/sec
S = Surface area of packing in sqft/cuft of packing. (see Table 18.5)
F = Fraction of void (see Table 18.5)
μ = Viscosity of liquid in cps.

Step 5. Solve the equation from step 4 to give a value for V. This is the superficial velocity of the vapor at flood. Designing for 80% of flood multiply V by 0.8.

Step 6. Divide the total cuft/sec of the vapor flowing in the tower by 0.8V to give the tower cross-sectional area in sqft. Calculate the tower diameter from this area.

Step 7. The next part of the calculation is to determine the height of the tower. The number of theoretical trays has been determined by either the fractionation or absorption calculation described in Chapter 3. It is now required to establish either the actual number of trays for a trayed tower or the height of packing in the case of a packed tower. This calculation deals with the second of these.

The next step sets out to establish the HETP which is the height equivalent to a theoretical tray.

Step 8. The HETP is calculated from the following equation:

$$\text{HETP} = K_1 \cdot G_{\text{h}}^{K_2} \cdot D^{K_3} \cdot \frac{62.4 \times \alpha \times \mu^2}{\rho_1}$$

where

$K_{1,2,3}$ = factors from Table 18.6
D = tower diameter in ins
α = relative volatility of the more volatile component in the liquid phase (see step 9 below)
μ = viscosity in cps
G_{h} = mass velocity of the vapor in lbs/hr sqft

Step 9. To determine the relative volatility α, select a key light component that is in the lean liquid and the wet gas. The relative volatility is the equilibrium constant of the lightest significant component in the rich liquid divided by the equilibrium constant of the light key component. Solve for a value of HETP and multiply this by the number of theoretical stages to give the total packed height.

Step 10. Determine the number of beds to accommodate the packed height. Allow space between the beds for vapor/liquid redistribution and holdup plates. Use Figure 18.9 as a guide for this. The tower height will be the sum of beds, internal distributors packing support trays, liquid hold up and vapor disengaging space.

An example calculation now follows.

Example calculation

In this example the number of theoretical trays for an absorption unit has been fixed as 4. The compositions of the "wet" gas and the lean liquid have been given and used to determine the composition and quantities of the rich liquid and the lean gas. The quantities to be used in the following calculation are as follows:

Rich liquid leaving the bottom of the absorber = 452.66 moles/hr
Wet gas entering the bottom of the tower = 1,018.35 moles/hr

Their respective composition and conditions are as follows:

Wet gas

	Mole Frac	Mole wt	Weight
H_2	0.467	2.0	0.93
C_1	0.190	16.0	3.40
C_2	0.059	30.0	1.77
H_2S	0.242	34.0	8.24
C_3	0.604	44.0	1.32
iC_4	0.006	58.0	0.35
nC_4	0.006	58.0	0.35
Total	1.000	16.0	16.00

$$\text{Temperature} = 95^\circ\text{F} \quad \text{Pressure} = 175\,\text{psia}$$

$$\text{Cuft} = \frac{378 \times 14.7 \times 555}{175 \times 520} = 33.89$$

$$\rho_v = 16/33.89 = 0.473\,\text{lbs/cuft.}$$

Rich liquid

	Mole Frac	Mole wt	wt Fact	lbs/Gal	Vol Fact
H_2	0.004	2.0	0.002	–	–
C_1	0.013	16.0	0.208	2.5	0.083
C_2	0.016	30.0	0.480	2.97	0.162
H_2S	0.092	34.0	3.128	6.56	0.477
C_3	0.028	44.0	1.232	4.23	0.291
iC_4	0.011	58.0	0.638	4.68	0.136
nC_4	0.013	58.0	0.754	4.86	0.155
C_9	0.823	128.0	105.344	6.02	17.499
Total	1.000		111.786	5.95	18.803

Temperature $= 95\ ^\circ$F

Sg @ 60 $= 0.715$
Sg @ 95 $= 0.696$
ρ_1 @ 95°F $= 43.3$ lbs/cuft.

$$\frac{L}{G}\left[\frac{\rho_v}{\rho_1}\right]^{1/2} = \frac{V^2 \cdot S \cdot \rho_v \cdot \mu^{0.3}}{g\, F^3\, \rho_1}$$

where

V = the vapor velocity at flood in ft/sec.
g = 32.2 ft/sec.
S = Surface area of packing in sqft/cuft of packing. (see Table 18.5) = 36 sqft/cuft
F = Fraction of void (see Table 18.5) = 0.79
μ = Viscosity of liquid in cps. = 0.56 cps

$$\frac{L}{G}\frac{[\rho_v]^{1/2}}{[\rho_1]} = \frac{50{,}670}{16{,}325} \times \left[\frac{0.473}{43.3}\right]^{1/2} = 0.324$$

From Figure 18.10 = 0.055.

Then:

$$\frac{V^2 \times S \times \rho_v \times \mu}{g \times F^3 \times \rho_1} = 0.055$$

$$\begin{aligned}
V^2 &= \frac{0.055}{0.022} = 2.5\\
V &= 1.58 \text{ ft/sec.}\\
\text{@ 80\% of flood } V &= 1.58 \times 0.8\\
&= 1.26 \text{ ft/sec.}\\
\text{Total vapor flow} &= 16{,}325 \text{ lbs/hr}\\
&= \frac{16{,}325}{0.473} = 34{,}514 \text{ cuft/hr.}\\
&= 9.59 \text{ cuft/sec}\\
\text{cross-sectional area} &= \frac{9.59}{1.26} = 7.6 \text{ sqft.}\\
\text{Tower diameter} &= 3.1 \text{ ft say } 3.25 \text{ ft or } 39''.
\end{aligned}$$

To calculate HETP. Use 2″ Berl Saddles.

$$\text{HETP} = K_1 \cdot G_H^{K_2} \cdot D^{K_3} \cdot \frac{62.4 \times \alpha \times \mu_1}{\rho_1}$$

where

$K_1 = 0.56$
$K_2 = -0.2$
$K_3 = 1.11$
α = relative volatility (neglect H_2 and C_1 in liquid composition as non-condensable). The Key component is C_3 then is KC_2/KC_3, which is $4.6/1.0 = 4.6$.
$G_H = 16{,}325/7.6 = 2{,}148$ lbs/hr·sqft
D = tower diam = 39″

$$\text{HETP} = \frac{0.56 \times 39 \times 62.4 \times 4.6 \times 0.56}{(2{,}148)^{0.02} \times 43.3}$$
$$= 25.83 \text{ ft per theoretical tray}$$

The number of theoretical trays was fixed at 4 for this separation. Then using 4 theoretical trays the total height of packing = 4 × 25.83 ft = 103 ft call it 100 ft. Five packed beds each 20 ft would satisfy the required duty. Using Figure 18.9 as a guide the tower height is developed as follows:

Bottom tan to HLL (holdup). Liquid is feed to a stripping column, therefore let the holdup time be 3 mins to NLL. Then NLL = 6.9 ft say 7.0 ft and HLL = 10 ft.

HLL to vapor inlet distributor. This will be set at 2.0 ft.

Distributor to bottom bed packing support. This will be set at 1.0 ft.

Bottom packed bed support to top of packed bed. Packed height which is 20 ft.

Top of bottom bed to bottom of next bed. Set this at 3.0 ft to allow for a liquid weir type distributor.

Height to top of top bed.

Packed height which is 4 × 20 ft = 80
4 distributors 12 ft
total = 92 ft

Top of top bed to top tan (tangent). Make this 5 ft to allow for liquid distribution tray and liquid inlet pipe.

Total height tan to tan = 130 ft.

Drums and drum design

Drums may be horizontal vessels or vertical. Generally drums do not contain complex internals such as fractionating trays or packing as in the case of towers. They are used however for removing material from a bulk material stream and often use simple baffle plates or wire mesh to maximize efficiency in achieving this. Drums are used in a process principally for:

- Removing liquid droplets from a gas stream (knockout pot) or separating vapor and liquid streams
- Separating a light from a heavy liquid stream (separators)
- Surge drums to provide suitable liquid hold up time within a process
- To reduce pulsation in the case of reciprocating compressors

Drums are also used as small intermediate storage vessels in a process.

Vapor disengaging drums
One of the most common examples of the use of a drum for the disengaging of vapor from a liquid stream is the steam drum of a boiler or a waste heat steam generator. Here the water is circulated through a heater where it is risen to its boiling point temperature and then routed to a disengaging drum. Steam is flashed off in this drum to be separated from the liquid by its superficial velocity across the area above the water level in the drum. The steam is then routed to a super-heater and thus to the steam main. The performance of the steam super-heater depends on receiving fairly "dry" saturated steam. That is steam containing little or no water droplets. The separation mechanism of the steam drum is therefore critical. The design of a vapor disengaging drum depends on the velocity of the vapor and the area of disengagement. This is expressed by the equation:

$$V_c = 0.157\sqrt{\frac{\rho_1 - \rho_v}{\rho_v}}$$

where
V_c = critical velocity of vapor in ft/sec
ρ_1 = density of liquid phase in lbs/cuft
ρ_v = density of vapor phase in lbs/cuft.

The area used for calculating the linear velocity of the vapor is:

- The vertical cross-sectional area above the high liquid level in a horizontal drum
- The horizontal area of the drum in the case of vertical drums

The allowable vapor velocity may exceed the critical, and normally design velocities will vary between 80% and 170% of critical. Severe entrainment occurs however above 250% of critical. Table 18.8 gives the recommended design velocities for the various services. The minimum vapor space above the liquid level in a horizontal drum should not be less than 20% of drum diameter or 12″, whichever is greater.

Table 18.8. Some typical drum applications

Service	Liquid surge and distillate	Settling drums	Compressor suction	Fuel gas KO drums	Steam drums	Water disengaging drums
			Cent Recip			
Allowable vapor velocity without CWMS % V_c	170		80 80	170	–	170
Allowable vap velocity with 1CWMS % V_c			150 120		100	
Allowable vap velocity with 2CWMS % V_c			– –		150	
Liquid hold up set by	Water settling	Settling Requirements	10 min liquid spill	Should be at least volume of a 20 ft slug of condensate.	1/3 the heater and steam piping volume	50 ins per minimum settling rate for Hydrocarbon vapors from water.
	Minimum instrument	Minimum instrument	When taking suction from absorbers	Following an absorber—5 mins on total lean oil circulation		Minimum height to low level 1.5 ft
	Controlling Process	Controlling process	For refrigerators—5 mins based on largest cooling unit			
	Inventory requirement					
Normal drum position	Horizontal	Horizontal	Vertical	Vertical	Vertical	Horizontal
Type of nozzle inlet	90° bend	90° bend	Tee Dist	Flush	Tee Dist	90° bend
Outlet vapor	Flush	–	Flush	Flush	Flush	Flush
Outlet liquid	Flush	Flush	Flush	Flush	Flush	Flush

Crinkled wire mesh screens (CWMS) screens are effective entrainment separators and are often used in separator drums for that purpose. When installed they improve the separation efficiency so vapor velocities much above critical can be tolerated. They are also a safeguard in processes where even moderate liquid entrainment cannot be tolerated.

CWMS are now readily available as packages that include support plates and installation fixtures. Normally for drums larger than 3 ft in diameter 6″ thick open mesh type screen is normally used.

Liquid separation drums
The design of a drum to perform this duty is based on one of the following laws of settling:

Stokes law

$$V = 8.3 \times 10^5 \times \frac{d^2 \Delta S}{\mu}$$

When the Re number is < 2.0.

Intermediate law

$$V = 1.04 \times 10^4 \times \frac{d^{1.14} \Delta S^{0.71}}{S_c^{0.29} \times \mu^{0.43}}$$

When the Re number is 2–500.

Newtons law

$$V = 2.05 \times 10^3 \times \left[\frac{d \Delta S}{S_c}\right]^{1/2}$$

When the Re number is > 500.

where

$$\text{Re number} = \frac{10.7 \times d \cdot V \cdot S_c}{\mu}$$

V = settling rate in ins per minute
d = droplet diameter in ins
S = droplet specific gravity
S_c = continuous phase specific gravity
ΔS = specific gravity differential between the two phases
μ = viscosity of the continuous phase in cps

The following may be used as a guide to estimating droplet size:

Lighter phase	Heavy phase	Minimum droplet size
0.850 SG and lighter	Water	0.008 ins.
Heavier than 0.850	Water	0.005 ins.

The holdup time required for settling is the vertical distance in the drum allocated to settling divided by the settling rate. Some typical applications of drums for this service are given in Table 18.8.

Settling baffles, are often used to reduce the holdup time and the height of the liquid level.

Surge drums
This type of drum, the calculation of holdup time and surge control has been described fully in Chapter 4.0 under "Control Systems".

Pulsation drums or pots
This type of drum will be described in some detail in Part 3 of this chapter in the section on reciprocating compressors.

An example calculation on drum sizing now follows.

Example calculation
It is required to provide the dimensions and process data for the design of a reflux drum receiving the hydrocarbon distillate, water, and uncondensed hydrocarbon vapor from a distillation column. Details of flow and drum conditions are as follows:

Vapor: 12,000 lbs/hr, 40 mole wt, 300 moles/hr.
Distillate product: 76,650 lbs/hr, Sg @ 100°F 0.682.
Reflux liquid: 61,318 lbs/hr, Sg @ 100°F, 0.682.
Water: 17,381.
Temperature of drum: 100°F
Pressure of drum: 30 psia.

The drum is to be a horizontal vessel located on a structure 45 ft above grade. The liquid product is to feed another fractionating unit and therefore requires a holdup time of 15 min between LLL and HLL. The vapor leaving the drum is to be routed to fuel gas via a compressor, therefore complete disengaging of liquid droplets is required. Complete separation of water from the oil is required. However as the water is routed to a de-salter separator from the drum separation of oil from the water is not critical.

In all probability the surge volume required by the product will be the determining feature of this design. Setting the liquid levels in the drum will depend on the

settling out of the water from the hydrocarbon phase. The design will be checked for satisfactory vapor disengaging.

The design

1.0 Calculating the surge volume for the distillate product.

$$\text{Holdup time} = 15 \text{ min.}$$

$$\text{Product rate} = \frac{76,650 \text{ lbs/hr}}{0.682 \times 62.2} = 1{,}807 \text{ cuft/hr}$$

$$\text{Holdup volume} = \frac{1{,}807 \times 15}{60} = 452 \text{ cuft.}$$

Then volume of liquid between HLL and LLL is 452 cuft. Let this be 60% of total drum volume. Then drum volume 452/.60 = 753 cuft.

Using a length to diameter ratio (L/D) of 3, diameter and length are calculated as follows:

$$753 \text{ cuft} = \frac{\pi \cdot D^2}{4} \times 3D$$

$$D = \sqrt[3]{\frac{753 \times 4}{3\eta}}$$

$$= 6.8 \text{ ft make it } 7.0 \text{ ft}$$

$$L = 3 \times 7.0 \text{ ft} = 21.0 \text{ ft.}$$

2.0 Calculating water settling rate.

Using "intermediate law" then:

$$V = 1.04 \times 10^4 \times \frac{d^{1.14} \Delta S^{0.71}}{Sc^{0.29} \times \mu^{0.43}}$$

$$V = \text{settling rate in ins/min.}$$

$$d = \text{droplet size in ins} = 0.008''$$

$$Sc = \text{Sg of continuous phase} = 0.682$$

$$Sw = \text{Sg of water} = 0.993$$

$$\Delta S = 0.311$$

$$V = 1.04 \times 10^4 \times \frac{0.004 \times 0.44}{0.895 \times 0.78}$$

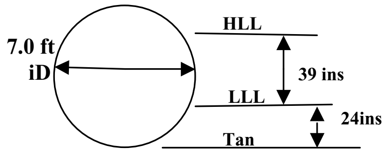
$$= 29.2 \text{ ins/min}$$

Check Re number:

$$\text{Re} = \frac{10.7 \times 0.008 \times 29.2 \times 0.682}{0.56}$$

$$= 3.0 \text{ so use of "intermediate law" is correct.}$$

3.0 Setting the distance between bottom tan and LLL



Sufficient distance or surge should be allowed below LLL to provide a LLL alarm at a point about 10% below LLL and bottom tangent. The remaining surge should be sufficient to provide the operator with some time to take emergency action (such as shutting down pumps).

Let LLL be 2 ft above bottom tan. Then the surge volume in this section is as follows:

R = 2/7 = 0.286. From Chapter 3 Appendix Figure A4.0 = 0.237 area of section = 0.237 × 38.48 = 9.1 sqft and volume = 21 × 9.1 = 191 cuft.

$$\begin{aligned}\text{Total flow rate} &= \text{Product} + \text{Reflux} + \text{Water.}\\ &= 3{,}531.4 \text{ cuft/hr} = 58.86 \text{ cuft/min.}\end{aligned}$$

Minutes of hold up below LLL = 3.25 mins

By the same calculation holdup after alarm = 2.9 mins, which is satisfactory.

4.0 Checking settling time for the water

At the LLL a distance of 2 ft from Tan
Residence time for liquid below LLL = 3.25 min.

Minimum settling time required:

$$\begin{aligned}&\frac{\text{Vert distance to bottom of drum}}{\text{Settling rate}}\\ &= \frac{24''}{29.2 \text{ inch/min}}\\ &= 0.82\,\text{min},\end{aligned}$$

which is adequate.

5.0 Calculating height of HLL above LLL

$$\begin{aligned}\text{Total volume to HLL} &= 191 + 452 = 643 \text{ cuft}\\ \text{Area above HLL} &= \frac{808 \text{ cuft} - 643 \text{ cuft}}{21 \text{ ft}}\\ &= 7.58 \text{ sqft.}\end{aligned}$$

Using table in Appendix of Chapter 3

$$\frac{A_D}{A_s} = \frac{7.58}{38.48} = 0.197 \quad R = 0.251$$

$$r = 0.251 \times 7.0 = 1.76 \text{ ft.}$$

$$\text{Height of HLL above LLL} = 7 - (1.76 + 2.0)$$
$$= 3.25 \text{ ft. (39 ins)}$$

6.0 Checking the vapor disengaging space.

$$V_c = 0.157\sqrt{\frac{\rho_l - \rho_v}{\rho_v}}$$

where

V_c = critical velocity of vapor in ft/sec
ρ_l = density of liquid phase in lbs/cuft = 42.42
ρ_v = density of vapor phase in lbs/cuft = 0.216

$$V_c = 2.28 \text{ ft/sec.}$$

Actual velocity of vapor is as follows:

Cross-sectional area of vapor space above HLL = 7.58 sqft

$$\text{Vapor linear velocity} = \frac{59{,}840 \text{ cuft/hr}}{7.58 \times 3{,}600} = 2.19 \text{ ft/sec}$$

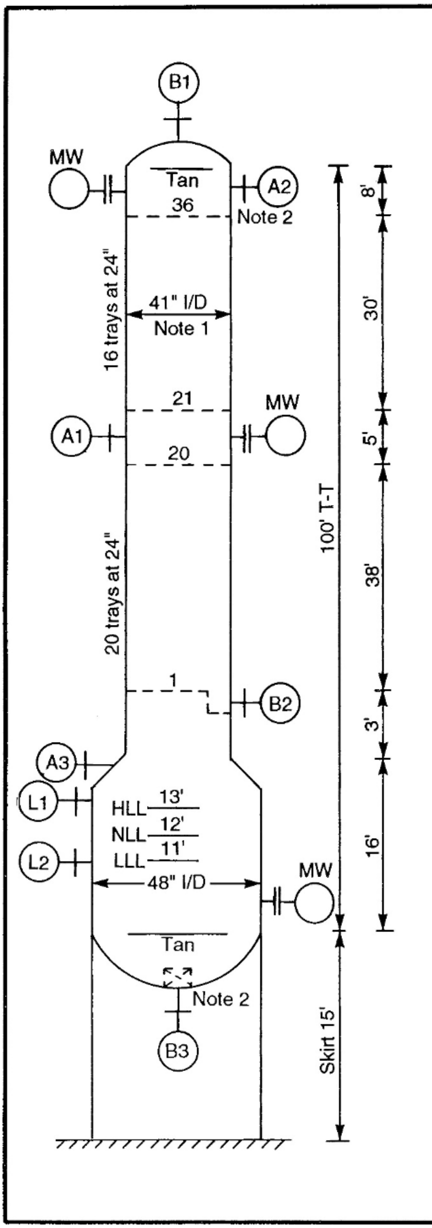
which is 96% of critical.

The drum design meets all necessary criteria and will be used.

Specifying pressure vessels

Process engineer's responsibility extends to defining the basic design requirements for all vessels. These data include:

- The overall vessel dimensions
- The type of material to be used in its fabrication
- The design and operating conditions of temperature and pressure
- The need for insulation for process reasons
- Corrosion allowance and the need for stress relieving to meet process conditions
- Process data for internals such as trays, packing, etc.
- Skirt height above grade
- Nozzle sizes, ratings and location (not orientation)



DESIGN CONDITIONS

Pressure + 237 psig at 490 °F
Vacuum - 7.0 psia at 345 °F
Low temp. - N/A °F at ____ psia
Max liquid level 13.5 ft
SG of liquid 0.805 at 60 °F

OPERATING CONDITIONS

Pressure + TOP 205 / BOT 212 psig at 158 / 440 °F
Vacuum - 7.0 psia at N/A °F
Low temp. - N/A °F at N/A psia

H_2 partial press 1.1 psia
at 158 °F

MATERIALS		CORR. ALLOW
Shell	Carbon steel	1/8"
Lining	None	—
Internal	Carbon steel	1/8"
Trays		1/8"
Cap \ valves		1/8"

DESCRIPTION	Bulk density	Δ *P* psi
Packing		
Catalyst	N/A	

Mist eliminator None

Insulation Hot 2" MIN. Cold N/A
Not req. for proc. –
Fireproofing Yes ____ No x

NOTES and SPECIAL CONDITIONS

Stress relieving Yes ____ No x
(1) Use standard 42" sched 'x' seamless steel pipe for fabrication
(2) Supply standard distributor and vortex breaker
(3) Attachment 1 — schedule of nozzles
Attachment 2 — tray data

Figure 18.11. A typical process data sheet for columns.

Typical process data sheets used for specifying columns and horizontal drums are given in Figures 18.11 and 18.12 (with their attachments) respectively. These data sheets have been completed to reflect the examples calculated in this chapter. The following paragraphs describe and discusses the contents of these data sheets.

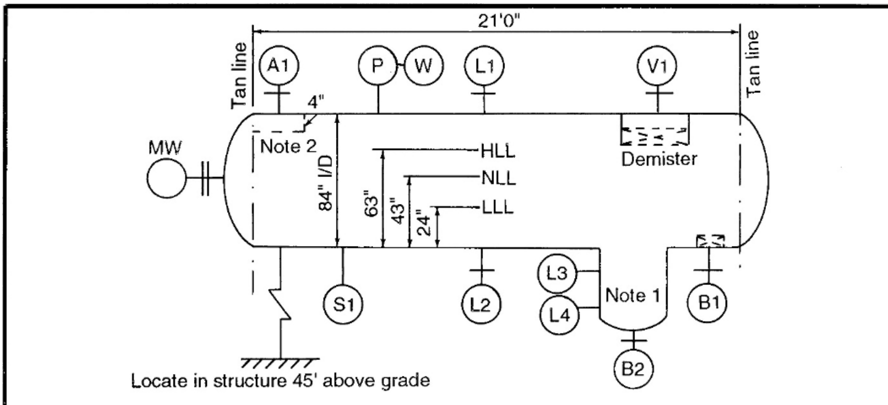
The attachments to Figure 18.11 are as follows:

Figure 18.11. Attachment 1. Nozzle Schedule

Ref	Description	Size, in inch	RTG
A 1	Feed inlet nozzle	6	150 RF
A 2	Reflux inlet nozzle	4	150 RF
A 3	Inlet from reboiler	6	300 RF
B 1	O/Head vapor outlet	8	150 RF
B 2	Outlet to reboiler	4	300 RF
B 3	Bottom product outlet	3	300 RF
L1 L2	Instrument nozzles	$^3/_4$	300 RF
MW	Manways	24	150 RF

Figure 18.11. Attachment 2. Tray Data Sheet

Vessel no	C401	
Vessel name	Reformate stabilizer	
Description of material	Un-stabilized light hydrocarbons	
Section	Top trays 21–36	Bottom trays 1–20
Total trays in section	16	20
Max ΔP per tray, psi	0.25	0.25
Conditions on tray	Top Tray No 30	Bottom Tray No 1
Vapor		
Temp, °F	167	440
Pressure, psig	205	212
Density, lbs/cuft	1.69	2.0
Rate, lbs/hr	47,700	71,021
ACFS	7.83	9.81
Liquid		
Temp, °F	162	430
Viscosity, Cps	0.3	0.85
Density, lbs/cuft	27.3	38.2
Mole weight	57	100
Rate, lbs/hr	33,273	104,950
Rate, cuft/min	20.34	45.79
Tower diameter, ft	3′ 5″	
Number of tray passes,	One	
Type of tray	Valve	
Tray spacing, ins	24	



DESIGN CONDITIONS	Pressure 50 psig H_2 Par press NIL psi Max liquid level 70 in.	Vacuum 7.5 psia Temperature 250 °F Design for steam out Yes x No
OPERATING CONDITIONS	Pressure 15 psig Normal liguid level 43 ins.	Temperature 100 °F
CORROSION ALLOWANCES	Shell $^1/_8$ in., water boot $^1/_4$ in.	
MATERIALS	Shell CS Lining (Note 3) Mist eliminator: Yes	Internals CS Packing None
Type of packing	None	
Type of mist eliminator	Wire mesh min, 4 in. thick	Insulation Yes No x Fireproffing Yes x No
SG of liquid(s)	0.682 at 100 °F	Stress relieve Yes No x

Notes and ref. drawings

1. Supply 36" T-T x 24" I/D boot as shown
2. Splash distribution 8" SQ x 4" deep
3. Gunnite line to HLL ($1^1/_2$" thick) shell only. No lining in boot

Figure 18.12. A typical process data sheet for horizontal vessels (drums).

The vessel sketch

This particular vessel is a light ends fractionator and has a single tray diameter (i.e., it is not swaged). The tower contains 36 valve trays on a 24″ tray spacing and has a calculated diameter of 39″ for the trayed section. This diameter will be specified as 41″ i/d however. This can be met by a standard 42″ schedule "X" pipe and this will reduce the cost of the vessel. The overall dimension for the tower is completed by setting the height of the tower from Tan to Tan. In the example here this has been done as follows:

$$\text{Height of trayed section} = (\text{No of trays} - 1.0) \times \text{tray spacing}.$$

The fractionation calculation (see Chapter 3.2 for this type of calculation) has determined 36 actual trays for this tower. Thus the trayed height is $(36 - 1) \times 24'' = 70$ ft.

Add another 3 ft to accommodate the feed inlet distributor on tray 20. Then total trayed height is 73 ft.

Bottom of the tower must accommodate the liquid surge requirement. As the tower diameter is relatively small a swaged section of 4 ft diameter will be considered below the bottom tray for surge. The liquid product goes to storage, therefore the surge requirement need not be more than 3 min on product.

From the unit's material balance the bottom product is as follows:

$$\begin{aligned} \text{Weight per hour} &= 117{,}513 \text{ lbs} \\ \text{Temperature} &= 440^\circ\text{F} \\ \text{Density @ } 440^\circ\text{F} &= 40 \text{ lbs/cuft} \end{aligned}$$

Then hot cuft/min of product = 48.96.

The product goes to storage, therefore only 2–3 min surge is required. This will be set at 3 min surge to NLL.

$$\begin{aligned} \text{Total surge to NLL} = 48.96 \times 3 &= 146.9 \text{ cuft.} \\ \text{Cross-sectional area of surge section} &= \eta/4 \times 4^2 \\ &= 12.6 \text{ sqft.} \\ \text{Height of NLL above Tan} &= \frac{146.9}{12.6} \\ &= 11.7 \text{ ft make it 12 ft.} \end{aligned}$$

level range will be 24″ then HLL will be 12 +1 = 13 ft above NLL.

Let the reboiler inlet nozzle be 3 ft above HLL = 16 ft above Tan.

Allow another 3 ft between reboiler inlet nozzle to bottom tray. This provides adequate space for vapor separation from the high liquid level.

At the top of the column space must be provided between the top tray and the top vapor outlet nozzle to accommodate the reflux return distributor and vapor disengaging from the top tray. Let this be 8 ft from top tray to the tower top Tan line. Then

Total tower height:

$$\begin{aligned} \text{From bottom Tan to Bottom tray} &= 19.0 \text{ ft} \\ \text{The trayed section} &= 73.0 \text{ ft} \\ \text{From top tray to top Tan} &= 8.0 \text{ ft} \\ \text{Total} &= 100.0 \text{ ft Tan to Tan} \end{aligned}$$

These overall dimensions are now inserted on the vessel diagram as shown in Figure 18.11. An attached sheet will be included to give the nozzle description, size and flange rating referring to those shown in the small circles on the sketch. A schedule of flange ratings for carbon steel is given in Table 18.9.

The only other dimension that will be shown on the sketch is that of the Skirt. Now the vessel is installed onsite supported by a metal skirt fixed to the concrete foundation of the vessel. The height of this skirt is fixed by the following criteria. If the product

Table 18.9. Schedule of flange ratings for carbon steel

Flange ratings psig Serv temp, °F	150	300 Max	400 operating	600 pressures	900 Psig
100	275	720	960	1,440	2,160
150	255	710	945	1,420	2,130
200	240	700	930	1,400	2,100
250	225	690	920	1,380	2,070
300	210	680	910	1,365	2,050
350	195	675	900	1,350	2,025
400	180	665	890	1,330	2,000
450	165	650	870	1,305	1,955
500	150	625	835	1,250	1,875
550	140	590	790	1,180	1,775
600	130	555	740	1,110	1,660
650	120	515	690	1,030	1,550
700	100	470	635	940	1,410
750	100	425	575	850	1,275
800	92	365	490	730	1,100
850	82	300	400	600	900
900	70	225	295	445	670
950	55	155	205	310	465
1,000	40	85	115	170	255

from the bottom of the tower is to be pumped (as is usual), the skirt height must be such as to accommodate the suction conditions for the pump. The most important of these conditions is the head required to meet the pump's net positive suction head (NPSH). See Part 2 of this chapter "PUMPS" for details. Usually a skirt height of 15 ft meets most NPSH requirements. The second consideration dictating skirt height is the head required by a thermosyphon reboiler. If the vessel is new and being designed the skirt height of 15 ft remains adequate with properly designed piping to and from the reboiler.

Design conditions

This particular tower operates at 212 psig and 440°F in the bottom and 205 psig & 158°F at the top. It is therefore classed as a pressure vessel and will be fabricated to meet a pressure vessel code. The most common of these codes is the ASME code either Section 1 or Section 8. Most vessels in the chemical industry are fabricated to ASME Section 8.When fabricated, inspected, and approved it will be stamped to certify that its construction conforms to this pressure vessel code.

Among the data required by the code and for complete vessel engineering and fabrication are the design conditions of temperature and pressure for the vessel.

Design pressure
The design pressure is based on the maximum operating pressure at which the relief valve will open plus a suitable safety increment. The following table provides a guide to this increment:

Maximum: Operating pressure, psig	Design pressure, psig
Full or partial vacuum	50
0–5	50
6–35	50
36–100	Operating + 15
101–250	Operating + 25
251–500	Operating + 10%
501–1,000	Operating + 50
over 1,000	Operating + 5%

In cases where vessels relieve to a flare header it may be necessary to add a little more to the differential between operating and design pressures to accommodate for the flare back pressure.

Design temperatures
The following table may be used as a guide to the max and min design temperatures. *Note:* Very often companies will have their own standards for these design criteria. The table given here may be used if there are no company standards.

Maximum: Operating temperature, °F	Design temperature, °F
Ambient –200	250
201–450	Operating + 50
Over 450	Divide into zones add 50 to each operating zone.
Vessels	
up to 225	250
226–600	Operating + 25
Over 600	Operating + 50
Minimum: Operating temperatures, °F	**Minimum: Design temperatures, °F**
15 to Ambient	Operating − 25
14 to 10	− 20
−10 to 80	Operating − 10
Below −80	Operating

A vacuum condition can exist in a tower during normal steam out if the tower is accidentally shut in and the steam valve closed. Normally a design vacuum pressure of 7 psia is specified at the steam saturated temperature to cover this contingency.

Low temperature
This applies to towers in cryogenic services (such as de methanizers, and LNG plants). There may be a situation in a non cryogenic service where rapid de-pressurizing causes sub zero temperatures to exist. If this is a situation that can exist for several hours and occurs frequently this condition should be entered. Otherwise make an appropriate remark in “Notes and Special Conditions”.

Max liquid level
This is the liquid level under operating conditions that will:

Either—Activate the high liquid level alarm.
Or—Shut down the feed pump.

whichever system is applicable to protect the plant operation. Usually this is quoted as the HLL and 1–2% of surge.

SG of liquid
Quote this as the SG of the liquid on which the surge volume was based. This SG is usually quoted at 60°F.

Operating conditions
In most fractionation towers there will be two distinctly different conditions of temperature and pressure—those for the tower top and those for the bottom of the tower. Both these conditions must be quoted in this case. The same situation may not necessarily arise in an absorption column.

Operating temperatures and pressures
Quote the calculated data as they will appear also on the process flow diagram. Show the tower top pressure and temperature first, followed by the bottom set of conditions. If the tower has been sized on data for more than one design case, show the highest numbers calculated for top and bottom. Also make a note in the "notes and Special Conditions" section of the cases the data was based on.

Other operating data
Vacuum conditions in this case only apply if the tower operates normally at sub atmospheric pressure. In this case quote the lowest pressure the tower will be operated on together with the normal operating temperature(s). Note in many vacuum fractionators there will be a spectrum of these conditions along the tower, these should be quoted for critical locations in the tower. Such locations would be feed inlet (flash zone), side stream and pump-around, draw off, tower top.

Low temperatures and the associated pressure apply only to cryogenic plants in this case.

Hydrogen partial pressure
This item is important to the metallurgist who will select the grade of metal to be used in the fabrication of the vessel. Generally the hydrogen partial pressure that will be quoted will be the one that exists at the tower top under normal operating conditions. For example the dew point calculation used in the sizing of the tower given in Figure 18.11 was based on the following tower top vapor composition.

	Mole fraction
H_2	0.005
C_1	0.021
C_2	0.117
C_3	0.378
iC_4	0.207
nC_4	0.268
iC_5	0.004
Total	1.000

The tower top pressure is 220 psia and the temperature is 158°F.

$$\text{Hydrogen partial pressure} = \frac{\text{Moles H}_2}{\text{Total Moles Gas}} \times \text{System Pressure}$$
$$= \frac{0.005}{1.0} \times 220 = 1.1 \text{ psia}$$

Materials and corrosion allowance

The process engineer will state the type of materials required to meet the process condition. For example where carbon steel only is to be used the process engineer indicates "CS". He is not normally required to state the grade of steel to be used, this is the responsibility of the vessel specialist or the metallurgist. However if the process engineer has a special knowledge of the material to be used and its specifics he should note it on the data sheet.

The same applies to the corrosion allowance. Normally 1/8″ is used for this allowance, however there maybe some mild corrosive condition existing which may justify using a higher number.

Description of internals

This is self explanatory when it refers to packed columns. In the case of trayed towers a separate data sheet giving sufficient data for tray rating and sizing is attached to the process data sheet front page. This is shown in the attachments to Figure 18.11.

Other common internals, such as distributors, vortex breakers, and the like are not normally shown on this data sheet. These are normally standard to a particular design office and will be added to the engineering drawings developed from this process data sheet later.

Insulation and fireproofing

The insulation requirement for heat conservation is specified by the process engineer as required. An approx thickness is shown. This will be checked later by the vessel specialist. In the case of fireproofing the process engineer indicates whether or not it is needed. The process engineer's relief valve sizing based on a fire condition takes into consideration the inclusion or not of fireproofing.

Notes and special conditions

This item is a "catch all" and is used to make note of whatever other information the process engineer may wish to add to the data sheet to ensure the equipment item will meet the process requirements. The question of stress relieving of the vessel is an item which is most important to the proper fabrication of the vessel and to its cost. The process engineer usually has knowledge whether this is needed or not to handle the process material at the conditions specified. He must therefore indicate this in this

section of the data sheet. Other entries in this item should be a list of the attachments to the data sheet.

Most of the process data used to define the requirements for a horizontal drum are the same as those applied to a column, and these have already been discussed for Figure 18.11. The data included in the example given in Figure 18.12 have been calculated earlier in this chapter. In the data sheet however a “boot” measuring 2 ft i/d × 3 ft high has been added to the outlet end of the vessel to accumulate the water phase for better control of its level and to allow the disengaging of the hydrocarbon from the water.

18.2 Pumps

Pumps in the petroleum and other process industries are divided into two general classifications which are

- Variable head-capacity
- Positive displacement

The variable head-capacity types include centrifugal and turbine pumps whilst the positive displacement types cover reciprocating and rotary pumps.

The centrifugal pump

Centrifugal pumps comprise a very wide class of pumps in which pumping of liquids or generation of pressure is effected by a rotary motion of one or several impellers. The impeller or impellers force the liquid into a rotary motion by impelling action, and the pump casing directs the liquid to the impeller at low pressure and leads it away under a higher pressure. There are no valves in centrifugal type pumps (except, of course, isolation valves for maintenance, etc.), flow is uniform and devoid of pulsation. Since this type of pump operates by converting velocity head to static head, a pump impeller operating at a fixed speed will develop the same theoretical head in feet of fluid flowing regardless of the density of the fluid. A wide range of heads can be handled. The maximum head (in ft of fluid) that a centrifugal pump can develop is determined primarily by the pump speed (rpm), impeller diameter and number of impellers in series. Refinements in impeller design and the impeller blade angle primarily effect the slope and shape of the head-capacity curve and have a minor effect on the developed head. Multistage pumps are available which will develop very high heads; up to 5,000 ft. and up to 120 gpm. This versatility in handling high pressure head makes the centrifugal pump the most commonly used type in the process industry.

The turbine pump

Turbine pumps are a type of centrifugal pumps designed to recover power in systems of high flow and high differential pressure. These pumps transmit some of the kinetic energy in the fluid into brake horsepower. The actual energy recovery is about 50% of the hydraulic horsepower available. This type of pump is expensive and is therefore not as widely used as the centrifugal pump.

The rotary pumps

Rotary pumps are positive displacement pumps. Unlike the centrifugal type pump these types do not throw the pumping fluid against the casing but push the fluid forward in a positive manner similar to the action of a piston. These pumps however do produce a fairly smooth discharge flow unlike that associated with a reciprocating pump. The types of rotary pumps commonly used in a process plant are:

The gear pump. This pump consists of two or more gears enclosed in a closely fitted casing. The arrangement is such that when the gear teeth are rotated they are unmeshed on one side of the casing. This allows the fluid to enter the void between gear and casing. The fluid is then carried around to the discharge side by the gear teeth, which then push the fluid into the discharge outlet as the teeth again mesh.

Screw pumps. These have from one to three suitably threaded screwed rotors of various designs in a fixed casing. As the rotors turn, liquid fills the space between the screw threads and is displaced axially as the threads mesh.

Lobular pumps. The Lobular pump consists of two or more rotors cut with two, three, or more lobes on each rotor. The rotors are synchronized for positive rotation by external gears. The action of these pumps is similar to that of gear pumps, but the flow is usually more pulsating than that from the gear pumps.

Vane pumps. There are two types of vane pumps: those that have swinging vanes and those that have sliding vanes. The swinging vane type consists of a series of hinged vanes which swing out as the rotor turns. This action traps the pumped fluid and forces it into the pump discharge. The sliding vane pump employs vanes that are held against the casing by the centrifugal force of the pumped fluid as the rotor turns. Liquid trapped between two vanes is carried around the casing from the inlet and forced out of the discharge.

Reciprocating pumps

These are positive displacement pumps and use a piston within a fixed cylinder to pump a constant volume of fluid for each stroke of the piston. The discharge from

reciprocating pumps is pulsating. Reciprocating pumps fall into two general categories. These are the simplex type and the duplex type. In the case of the simplex pump there is only one cylinder which draws in the fluid to be pumped on the back stroke and discharges it on the forward stroke. External valves open and close to enable the pumping action to proceed in the manner described. The duplex pump has a similar pumping action to the simplex pump. In this case however there are two parallel cylinders which operate on alternate stroke to one another. That is, when the first cylinder is on the suction stroke the second is on the discharge stroke.

Reciprocating pumps may have direct acting drives or may be driven through a crankcase and gear box. In the case of the direct acting drive the pump piston is connected to a steam drive piston by a common piston rod. The pump piston therefore is actuated by the steam piston directly. Reciprocating pumps driven by electric motors, turbines, etc are connected to the prime mover through a gearbox and crankcase.

Other positive displacement pumps

There are other positive displacement pumps commonly used in the process industry for special services. Some of these are:

Metering or proportioning pumps. These are small reciprocating plunger type pumps with an adjustable stroke. These are used to inject fixed amount of fluids into a larger stream or vessel.

Diaphragm pumps. These pumps are used for handling thick pulps, sludge, acid or alkaline solutions, and fluids containing gritty solid suspensions. They are particularly suited to these kind of service because the working parts are associated with moving the diaphragm back and forth to cause the pumping action. The working parts therefore do not come into contact with this type of fluid which would be harmful to them.

Characteristic curves

Pump action and the performance of a pump are defined in terms of their *Characteristic Curves*. These curves correlate the capacity of the pump in unit volume per unit time versus discharge or differential pressures. Typical curves are shown in Figures 18.13–15. Figure 18.13 is a characteristic curve for a reciprocating simplex pump which is direct driven. Included also is this reciprocating pump on a power drive.

Figure 18.14 gives typical curves for a rotary pump. Here the capacity of the pump is plotted against discharge pressure for two levels of pump speed. The curves also show the plot of brake horsepower versus discharge pressure for the two pump speed levels.

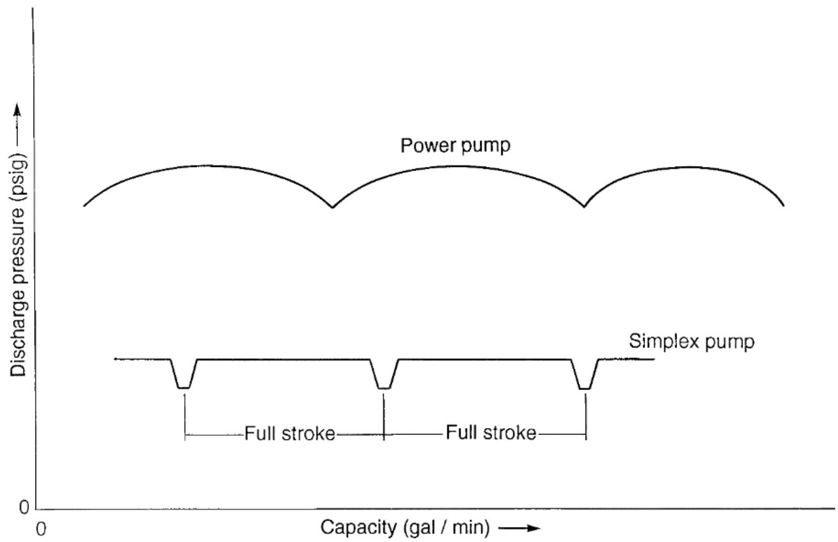


Figure 18.13. Characteristic curves for a reciprocating pumps.

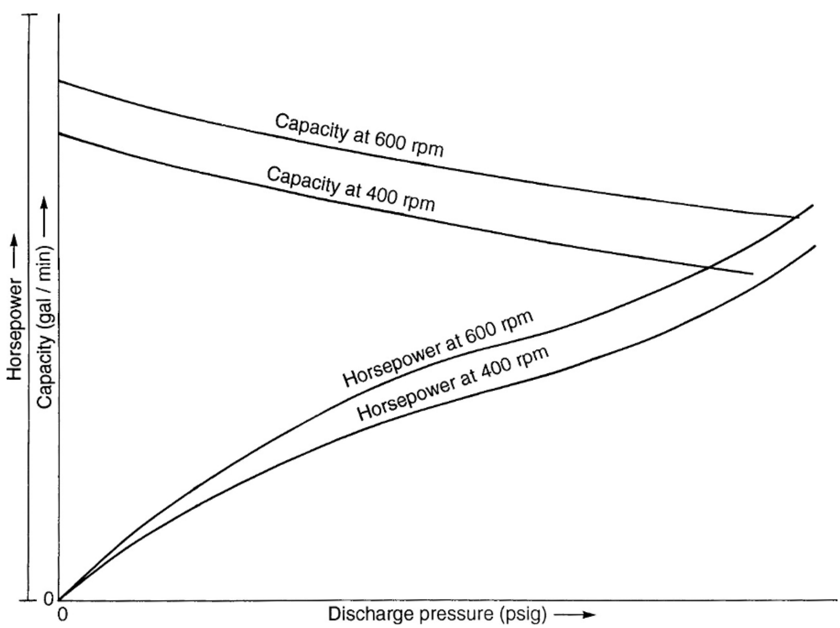


Figure 18.14. Characteristic curves for a rotary pump.

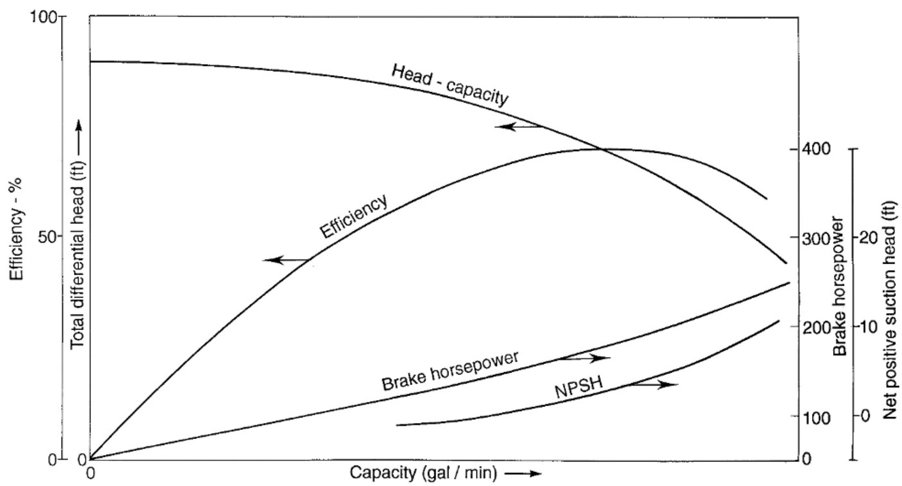


Figure 18.15. Characteristic curves for a centrifugal type pump.

Figure 18.15 is a typical characteristic curve for a centrifugal pump. This curve usually shows four pump relationships in four plots. These are:

- A plot of capacity versus differential head. The differential head is the difference in pressure between the suction and discharge
- The pump efficiency as a percentage versus capacity
- The brake horsepower of the pump versus capacity
- The net positive head (NPSH) required by the pump versus Capacity. The required NPSH for the pump is a characteristic determined by the manufacturer

Pump selection

Most industrial pumping applications favor the use of centrifugal pumps. The prominence of this type of pump stems from its ability to handle a very wide spectrum of fluids at a large range of pumping conditions. It is fitting that in considering pump selection the first choice has to be the centrifugal pump and all others become a selection by exception. Centrifugal pumps are generally the simplest in construction, lowest initial cost and simplest to operate and to maintain. This item therefore begins with the selection characteristics of the centrifugal type pump.

The centrifugal pump

Before looking at the selection of the centrifugal pump it is necessary to define the following terminology associated with pumps in general. These are:

- Capacity
- Differential Head
- Available NPSH
- Required NPSH

Capacity. This can be defined as the amount of fluid the pump can handle per unit time and at a differential pressure or head. This is usually expressed as gallons per minute at a differential head of so many pounds per square inch or so many feet.

Differential head. This is the difference in pressure between the suction of the pump and the discharge. It is usually expressed as PSI and FEET in specifying a pump. The following formula is the conversion from PSI to Feet:

$$\Delta H = \frac{2.31 \times \Delta P}{\mathrm{SG}}$$

where

H = the differential head in feet of fluid being pumped.
P = the differential pressure of the fluid across the pump measured in pounds per square inch.
SG = the specific gravity of the fluid at the pumping temperature.

Available NPSH. The available NPSH is the static head available (in feet or meters) above the vapor pressure of the fluid at the pumping temperature. This is a feature of the design of the system which includes the pump.

Required NPSH. Is the static head above the vapor pressure of the fluid required by the pump design to function properly. The required NPSH must always be less than the available NPSH.

Selection characteristics

Selection of any pump must depend on its ability to handle a particular fluid effectively, and the efficiency of the pump under normal operating conditions. The second of these primary requirements can be determined by the pump's characteristic curves. These have already been described earlier in this part of the chapter, and a further discussion on these now follows:

Capacity range

Normal

Figures 18.16 and 18.17 show the normal capacity range for various types of centrifugal pumps in two different speed ranges, 3,550 rpm, 2,950 rpm.

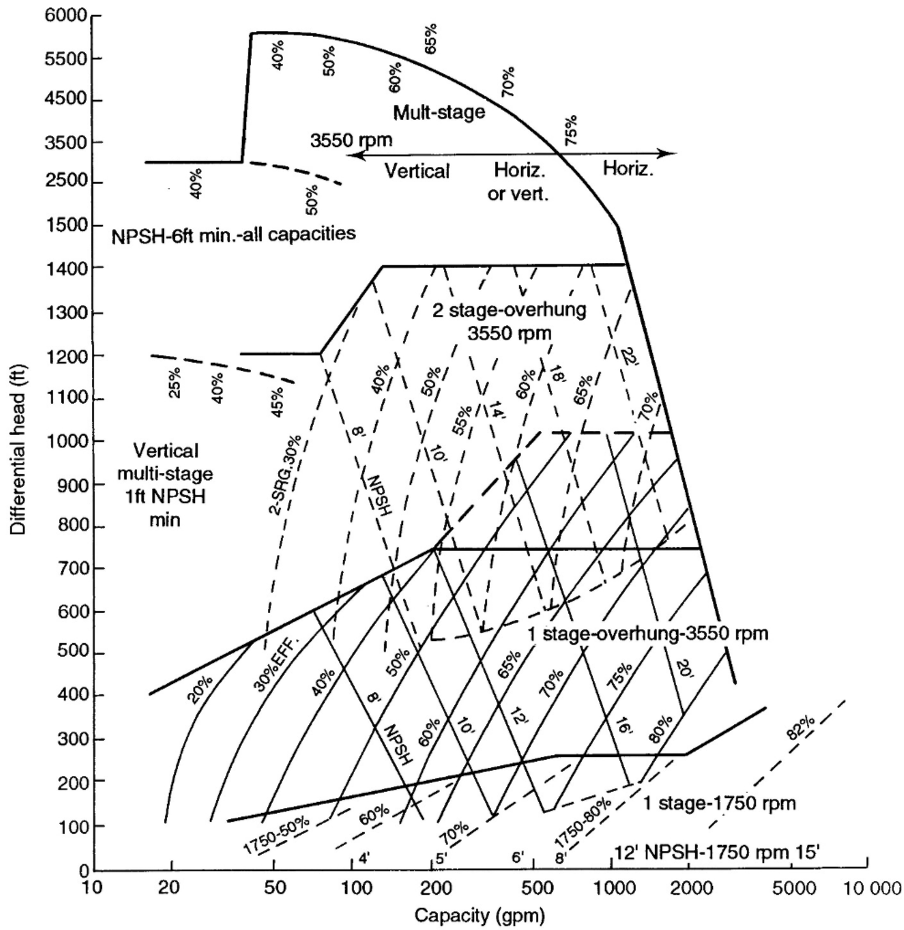


Figure 18.16. Centrifugal pumps at 3,550 rpm.

These values correspond to motor full load speeds available with current at 60 and 50 cycles, respectively. Most process applications call for these speed ranges. Lower speeds are for low or medium head and high capacity requirements, and for special abrasive slurries or corrosive liquids. Low capacity centrifugal pump applications may require special recirculation provisions in the process design to maintain a minimum flow through the pump. Because of practical consideration in impeller construction, the smallest available process type centrifugal pumps are rated at about 50 gpm.

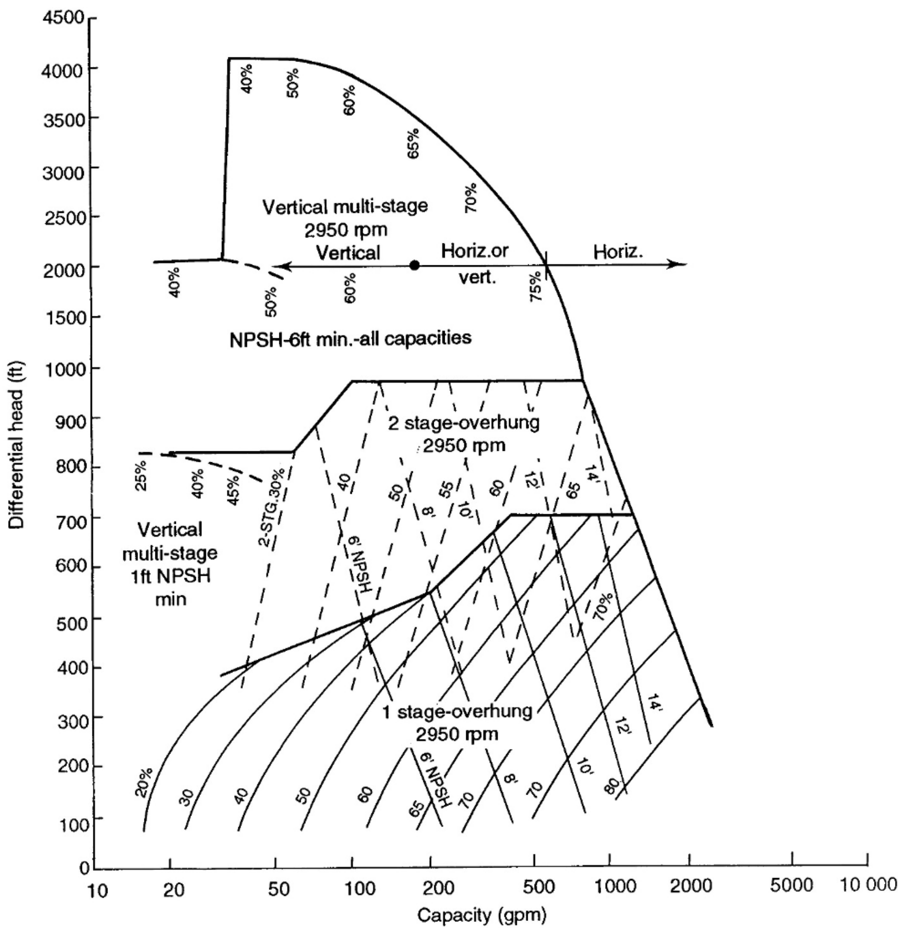


Figure 18.17. Centrifugal pumps at 2,950 rpm.

High and low capacity ranges

Pumps above the limits shown in Figure 18.16 and 18.17 will normally require large horsepower drivers. Special investigation of efficiency, speed, NPSH requirements, etc., will normally be justified. As an example, heads at or above the limits shown for multistage pumps at standard motor speed may be obtained by speed increasing gears (motor drive) or turbines to give pump operating speeds above maximum motor speeds, (NPSH requirements increase with speed).

In general, centrifugal pumps should not be operated continuously at flows less than approximately 20% of the normal rating of the pump. The normal rating for the pump is the capacity corresponding to the maximum efficiency point. The following

table lists minimum desirable flow rates which should be maintained by continuous recirculation, if the required process flow conditions are of lower magnitude:

Head range feet	Pump type	Minimum continuous capacity rating, gpm	Normal rating of pump, gpm
60 Cycle speed (3,550 rpm)			
To 100	1 stg	10	60
100–350	1 stg	15	75–100
350–650	2 stg	30	150
650–1,100	2 stg	40	160
400–1,200	Multistg	15	50
1,200–5,500	Multistg	40	100–120
50 Cycle Speed (2,950 rpm)			
To 75	1 stg	10	50
75–250	1 stg	15	60–80
250–450	2 stg	25	120
450–775	2 stg	30	130
250–850	Multistg	10	40
850–3,800	Multistg	30	80–100

Care must be exercised in the design of any recirculation system to insure that the recirculated flow does not increase the temperature of pump suction and cause increased vapor pressure and reduction of available NPSH.

For low head pumps that can operate at 1,750 or 1,450 rpm, the above normal and minimum continuous capacities are reduced by 50%.

Effect of liquid viscosity

When suitably designed, centrifugal pumps can satisfactorily handle liquids containing solids, dirt, grit and corrosive compounds. Though fluids with viscosities up to 20,000 SSU (440 centistokes) can be handled, 3,000 SSU (650 centistokes) is usually the practical limitation from an economical operating standpoint.

Effect of suction head

An important requirement is that there be sufficient NPSH at the eye of the first stage impeller. This is static pressure above the vapor pressure of the fluid handled to prevent vaporization at the impeller eye. Flashing of the fluid produces a shock or cavitation effect at the impeller which results in metal loss, noise, lowered capacity and discharge pressure, and rapid damage to the pump. NPSH requirements for various centrifugal pumps will normally vary from 6 to over 20 (in feet of fluid) depending on type, size

and speed. Vertical pumps can be built for practically no NPSH at all at the nozzle. These will have extended barrels in order to provide the required NPSH at the eye of the first stage impeller.

Efficiency

The efficiency of centrifugal pumps varies from about 20% for low capacity (< 20 gpm) pumps to a range of 70–80% for high capacity > 500 gpm pumps. Extremely large capacity pumps (several thousand gpm) may have efficiencies up to nearly 90%.

The rotary pumps

Rotary pumps deliver constant capacity against variable discharge pressure. This is a feature for all positive displacement pumps. Rotary pumps are available for process application over a range of 1 to 5,000 gpm capacity and a differential pressure up to about 700 psi. Displacement of the pump varies directly as the speed except in the case where the capacity may be influenced by a the viscosity of the fluid. In this case thick viscous liquids may limit capacity because they cannot flow into the cylinder fast enough to keep it completely filled. Rotary pumps are used mostly in low capacity service where the efficiency of a centrifugal pump would be very low.

Reciprocating pumps

The liquid discharge from reciprocating pumps is pulsating. The degree of pulsation is higher for SIMPLEX pumps than for the DUPLEX type. The pulsation is also higher for direct driven pumps than for those driven by motor or turbine through a gear box.

Pulsation is generally not a problem in the case of small low speed pumps of this type. It affects only the associated instrumentation which can be compensated for by local dampeners. However as the pump speed is increased the pulsation effect becomes more serious affecting the piping design of the system. Under these conditions of high pulsation instantaneous piping pressures may often exceed the design pressure of the piping. The piping must then be designed to meet this higher pressure requirement which invariably results in a higher cost. As an alternative, discharge dampeners are often considered but these too add to the cost of the pump installation.

Reciprocating pumps are used mostly in situations where their low piston speeds will withstand corrosive and abrasive conditions. They are ideal also for pumping at low

capacity against high differential head and where it is necessary to maintain a constant flow rate against a gradually increasing discharge pressure.

Evaluating pump performance

Many process engineers are involved with the day to day operation of process plants. Much of their duties in this respect is concerned with maintaining plant efficiency, locating trouble areas and solving operational problems. This Item is directed to these engineers and presents the calculation methods to check pump performance in terms of the pump horsepower and the available NPSH for a pump.

Brake horsepower efficiency

Actual running efficiency can be calculated from plant data. This may be compared with a typical expected efficiency to evaluate the pump performance. The steps below are followed to arrive at this efficiency figure.

Step 1. Obtain flow rate from plant readings. Also read discharge pressure and if available suction pressure. If this later reading is not available calculate it from source pressure, height of liquid above pump suction and frictional loss.

Step 2. Read stream temperature and obtain Sg of stream from lab data. Calculate Sg at flow conditions.

Step 3. Calculate differential head which is discharge pressure PSIA—suction pressure PSIA. Convert to differential head in feet by

$$\frac{\Delta P \times 144}{62.2 \times \text{Sg (at flow cond)}}$$

Step 4. Convert feed rate to pounds per hour. Then calculate hydraulic horsepower from the expression

$$\frac{\text{Head (ft)} \times \text{rate (lbs/hr)}}{\text{60 (mins)} \times \text{33,000 (ft-lbs/min)}}$$

Step 5. From motor data sheet obtain pump motor running efficiency from plant data read power usage in kW.

Step 6. Convert pump power to HP by dividing kW by 0.746. Multiply this by pump efficiency expressed as a fraction. This is the brake horsepower.

Step 7. Divide hydraulic horsepower by brake horsepower and multiply by 100 to give efficiency as a percentage.

Step 8. Check against Figure 18.16 or 18.17 to evaluate pump performance.
That is, the calculated efficiency within a reasonable agreement with the expected efficiency given by Figure 18.16 or 18.17. Should there be a large discrepancy then the appropriate mechanical or maintenance engineers should be informed.

Checking available NPSH

This needs to be done if the pump is showing signs of vibration and losing suction under normal operations.

Step 1. Obtain details of the fluid being pumped (temperature, Sg, flow rate, source pressure, and vapor pressure of fluid).
Step 2. Calculate the frictional pressure drop in the suction line.
Step 3. Calculate the suction pressure of the pump by taking source pressure and adding in the static head. For this calculation take static head as being from the bottom of the vessel (not from the liquid level).
Step 4. From the suction pressure calculated in Step 3 take out loss through friction.
Step 5. Calculate NPSH available as being net suction pressure less vapor pressure. This is usually quoted in feet (or meters) so convert using:

$$\frac{\text{PSI} \times 144}{62.2 \times \text{SG}}$$

Step 6. Check against manufacturer's data sheet for required NPSH. If the available NPSH is less than required the pump will continue to cavitate. Fill tank to a level that the vibration stops and maintain it at that level.

If the problem is really troublesome and maintaining a liquid level as suggested in step 6 above is not practical contact the pump manufacturer. Very often he is able to make some minor changes to the pump design that will solve the problem.

Example calculation
Example calculation No. 1—Pump brake HP efficiency.

$$\begin{aligned}\text{Flow capacity of pump at flow conditions} &= 200 \text{ gpm}\\ \text{(from plant data)} &= 82{,}632 \text{ lbs/hr}\end{aligned}$$

$$\begin{aligned}\text{Suction pressure (plant data)} &= 120 \text{ psig}\\ \text{Discharge pressure} &= 450 \text{ psig}\\ \Delta\text{Pressure} &= 330 \text{ psi}\\ \text{Sg at flow conditions} &= 0.827\\ \text{Differential head ft} &= \frac{330 \times 144}{62.2 \times 0.827} = 924 \text{ ft}\\ \text{Hydraulic horsepower} &= \frac{82,632 \times 924}{60 \times 33{,}000}\\ &= 38.56\end{aligned}$$

From motor rating, motor efficiency is 92%
Plant readings show motor power usage $= 48.1$ kW

$$\text{Then motor HP input} = \frac{48.1}{0.746} = 64.5 \text{ HP}$$
$$\text{Motor output} = 64.5 \times 0.92$$
$$= 59.3$$

This is brake HP

$$\text{Then pump efficiency} = \frac{38.56 \times 100}{59.3} = 65\%$$

Motor is 50 cycle speed. From Figure 4.5 for multistage 2,950 rpm. This efficiency figure is about right.

Example calculation No. 2—Checking available NPSH
Fluid is gas oil from surge drum at 250°F and 15 psig drum is 12 ft above grade (bottom of drum which is horizontal).

Boiling point at 34 psia is 488°F.

Vapor pressure of gas oil at 250°F = 0.8 psia (VP curves)

Source pressure	=	29.7 psia
Sg @ conditions	=	0.815
Head above grade	=	12 ft (to bottom of drum)
Pump center line	=	2 ft above grade
Liquid head to pump	=	10 ft = 3.5 psi
Friction pressure drop	=	0.4 psi (calculated from 0.2 psi/100)
Less friction	=	0.4
Less vapor press	=	0.8

$$\text{Available NPSH} = 33.2 - (0.4 + 0.8) = 32 \text{ psia}$$
$$= 91 \text{ ft}$$

Most pumps only require 10 ft or less.

Specifying a centrifugal pump

In all engineering companies and in most petroleum refineries two disciplines are responsible for correctly specifying a pump. These are the Mechanical Engineer and the Process Engineer. Most pumps are designed and built in accordance with set and accepted industrial codes, such as the API codes. The mechanical engineer ensures that the mechanical data supplied to the manufacturer for a particular pump meets the requirements of the code and standards to which the pump is to be built. The

process engineer develops and specifies precisely the performance required of the pump in meeting the process criteria of the plant. To accomplish this the mechanical engineer develops a mechanical specification, and the process engineer initiates the pump specification sheet.

The mechanical specification

This specification is in a narrative form and will contain at least the following topics:

- Scope—Introductory paragraph which gives the code the pump manufacturer is to conform to (such as API 610).
  A list of other standards the pump shall conform to, if these are required.
- Main Body of the Specification—This covers all additions and any exceptions to the selected code. It provides for the type of drive shaft acceptable if different to code. Items such as impeller size as a percent of maximum allowable by code is given. The need for special bearing arrangements in the case of multi stage pumps are detailed in this document.
- Ancillary equipment and piping arrangements—The specification describes in detail the type of cooling medium that shall be used. It provides a guide also to the piping requirements that is required to satisfy the cooling system(s).
- Seal or Packing requirements—The mechanical specification details the type of seal or packing that will be installed. It also provides details of the seal arrangement required if this is different to the standard code.
- Pump Mounting—Some installation guide is provided by this specification. The method by which the pump is mounted on the base plate is detailed. It also details under what conditions the manufacturer is to provide pedestal cooling facilities.
- Metallurgy—Although the process engineer will specify the general material of construction for the pump(such as carbon steel or cast iron etc.) it is the mechanical engineer who details this. This detail includes the specific grade of the material and in many cases its pre-operational treatment.
- Inspection—The mechanical specification will provide details of the inspection that the company will carry out during the manufacture of the pump and before its delivery. This will include dimensional checking during manufacture and some checks on the metallurgy. Prior to shipping the purchaser may require a running test of the pump and will witness this test. For this purpose the pump is run in the workshop under specified process conditions.

The mechanical specifications may continue to detail other requirements that the purchaser may wish. Its objective is to ensure that the pump when delivered is mechanically robust, is safe and easily maintainable. The mechanical specification must also be aware of the cost implication of the requirements on the pump, and to keep them as low as possible.

The process specification

The data provided by the process engineer must be sufficient to ensure that the pump delivered for the process purpose will meet the duty required of it. These data are furnished to the pump manufacturer in the form of a data sheet similar to the one shown here as Figure 18.18. The data sheet collects the essential input from the process engineer, the mechanical engineer and, later, by the manufacturer to describe fully what is required of the pump and what the manufacturer has supplied. All the data given here will be unique to this pump.

The process input to the pump specification shown on Figure 13.6 are those items marked with the 'P'. Input by other disciplines and the manufacturer are not indicated on the form. The process engineer compiles much of these data from an 'Hydraulic Analysis' of the piping system. A calculation sheet given as Figure 18.19 shows the development of this and is described as follows:

Compiling the pump calculation sheet

The pump number, title, and service

This first section of the calculation sheet is important because it identifies the pump and what it is intended to do. The item number and service description will be unique to this item and will remain as its identification throughout its life. All the data below this section will refer only to this pump and to no other. The item number may contain the suffix 'A', 'B', 'C', etc. This indicates identical pumps in parallel service or as spare or both. This section also shows how many of these pumps are motor driven and how many are turbine (steam) driven. Usually spare pumps in critical service will be turbine driven. The remark column in this section should give any information that will be of benefit to the pump manufacturer or future operators of the pump. For example, if the spare pump is turbine driven, the process engineer may require an automatic start up of the turbine on a 'low flow' of the pumped stream. This should be noted here.

Operating conditions for each pump

The details of the fluid to be pumped and a summary of the calculations given below are entered here. Starting on the left of this section:

Liquid: This is a simple definition of the pumped material. In the example given here this will simply be 'vacuum gas oil'.

Pumping temperature (PT): This is the temperature of the gas oil at the pump. There are two temperatures called for 'normal' and 'max'. The normal temperature is that shown

SHEET NO. P REV. ____
JOB NO. P DATE P
BY P CHK'D. ____
P.O. NO. ____

NOTE: ○ INDICATES INFORMATION TO BE COMPLETED BY PURCHASER;
□ BY MANUFACTURER

FOR P SITE P
UNIT P SERVICE P
NO. PUMPS REQ'D P NO. MOTORS REQ'D P ITEM NO. P PROVIDED BY ____ MTD BY ____
NO. TURBINES REQ'D P ITEM NO. P PROVIDED BY ____ MTD BY ____
PUMP MFR ____ SIZE AND TYPE (P-Type only) SERIAL NO. ____

OPERATING CONDITIONS, EACH PUMP

LIQUID P m^3/h at PT. NOR. P RATED P
____ DISCH. PRESS., kg/cm^2g P
PT, °C NOR. P MAX. P SUCT. PRESS., kg/cm^2g MAX. P RATED P
SP.GR. at PT P DIFF. PRESS., kg/cm^2 P
VAP. PRESS. at PT, kg/cm^2a P DIFF. HEAD, m P
VIS. at PT, Ssu P CP P NPSHA, m P
CORR/EROS. CAUSED BY P HYD. HP(metric) P

PERFORMANCE

PROPOSAL CURVE NO. ____
RPM ____ NPSHR (WATER) m ____
EFF. ____ metric BHP RATED ____
MAX. metric BHP RATED IMP ____
MAX. HEAD RATED IMP m ____
MIN. CONTINUOUS m^3/h ____
ROTATION (VIEWED FROM CPLG END) ____

CONSTRUCTION

NOZZLES	SIZE	RATING	FACING	LOCATION
SUCTION				
DISCHARGE				

CASE-MOUNT: □ CENTERLINE □ FOOT □ BRACKET □ VERT. (TYPE) ____
-SPLIT: □ AXIAL □ RAD; TYPE VOLUTE □ SGL □ DBL □ DIFFUSER
-PRESS: □ MAX. ALLOW. ____ kg/cm^2g ____ °C, □ HYDRO TEST ____ kg/cm^2g
-CONNECT: □ VENT □ DRAIN □ GAGE
IMPELLER DIA.: □ RATED ____ □ MAX. ____, □ TYPE: ____
MOUNT: □ BETWEEN BRGS □ OVERHUNG
BEARINGS-TYPE: □ RADIAL ____ □ THRUST ____
LUBE: □ RING OIL □ FLOOD □ OIL MIST □ FLINGER □ PRESSURE
COUPLING: □ MFR ____ □ MODEL ____
DRIVER HALF MTD BY: ○ PUMP MFR ○ DRIVER MFR ○ PURCHASER
PACKING: □ MFR & TYPE ____ □ SIZE/NO. OF RINGS ____
MECH. SEAL: □ MFR & MODEL ____ API CLASS. CODE ____
□ MFR CODE ____

SHOP TESTS

○ NON-WIT. PERF. ○ WIT. PERF.
○ NON-WIT. HYDRO ○ WIT. HYDRO
○ NPSH REQ'D. ○ WIT. NPSH
○ SHOP INSPECTION
○ DISMANT. & INSP. AFTER TEST
○ OTHER ____

MATERIALS

PUMP: CASE/TRIM CLASS ○ ____
BASEPLATE: □ ____

AUXILIARY PIPING

○ C.W. PIPE PLAN ____ ○ CU; ○ S.S.; ○ TUBING; ○ PIPE ____
□ TOTAL COOLING WATER REQ'D, m^3/h ____ ○ SIGHT F.I. REQ'D ____
○ PACKING COOLING INJECTION REQ'D: □ TOTAL m^3/h □ kg/cm^2g
○ SEAL FLUSH PIPE PLAN ____ ○ C.S. ○ S.S. ○ TUBING ○ PIPE ____
○ EXTERNAL SEAL FLUSH FLUID ____ □ m^3/h ____ □ kg/cm^2g ____
○ AUXILIARY SEAL PLAN ____ ○ C.S. ○ S.S. ○ TUBING ○ PIPE ____
○ AUX. SEAL QUENCH FLUID ____

VERTICAL PUMPS

PIT OR SUMP DEPTH ○ ____
MIN. SUBMERGENCE REQ'D. □ ____
COLUMN PIPE: □ FLANGED □ THREADED
LINE SHAFT: □ OPEN □ ENCLOSED
BRGS: □ BOWL ____ □ LINE SHAFT ____
BRG. LUBE □ WATER □ OIL □ GREASE
FLOAT & ROD ○ C.S. ○ S.S. ○ BRZ ○ NONE
FLOAT SWITCH □ ____
PUMP THRUST, kg □ UP ____ □ DOWN ____

MOTOR DRIVER

HP(metric) ____, RPM P FRAME ____ VOLTS/PHASE/CYCLES P
MFR ____ BEARINGS ____ LUBE ____
TYPE ____ INSUL ____ FULL LOAD AMPS ____
ENC ____ TEMP RISE, C ____ LOCKED ROTOR AMPS ____
○ VHS ○ VSS VERT. THRUST CAP., kg ____ MTR. ITEM NO. ____

APPROX. WT. PUMP & BASE ____
MOTOR ____ TURBINE ____

API STANDARD 610 GOVERNS UNLESS OTHERWISE NOTED. APPLICABLE TO: PROPOSALS ○ PURCHASE ○ AS BUILT ○

P = Specified by Process

Figure 18.18. A centrifugal pump specification sheet.

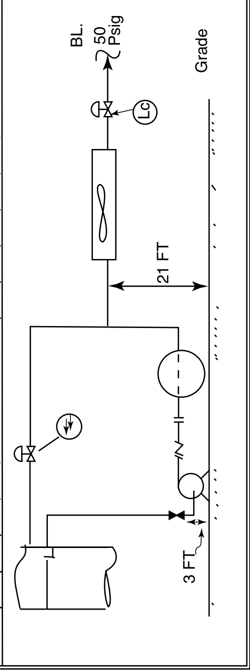
PUMP CALCULATION.

Item No P 103- A- Unit Crude Vacume Unit Sheet No 1 Rev 0
Service HGO Product and BPA Motor Drive 1
Turbine Drive 1 Remarks Turbine to have Auto Start By PSTT App. J.S

OPERATING	CONDITIONS (Each Pump)	TURBINE CONDITIONS
Liquid VAL GAS OIL	US GPN &Pt.Min ___MOR 13Q Rated 1585	Inlet Stean psig 600
PTP NOR 545 MAX 740	Dish press Psig 85.5	Temp P 670
SP GR & PT 0.755	Suct Press Psig Max 50 Rated –0?	Exhaust Psig 50
Vap Press & Pt.Psig 0.29	Diff Press Psi 86.2	PUMP MATERIALS
Vis & PT Cp 0.906	Diff Bead FT 264.3	Casing C.S.
Corrosion\Erosion None	NPSH Available, P 740 Ayd HP 79.7 (1)	Internal Parts C.S.

ALTERNATES		B.L		
DESTINATION :				
Destination Press	psig	50		
Static Head	psi	5.4		
Line Loss	psi	10.7		
Meter Loss	psi	0.2		
11-E-9/10Δ Ht Exchangers	psi	14.0		
Δ Control Valves	psi	5.20		
TOTAL DISCHARGE PRESS	psig	85.50		
SUCTION.				
Surce Press	psig	–14.4		
↓ Static Head	psi	14.7		
- Ststem Losses	psi	1.0		
TOTAL SUCTION PRESS	psig	–0.7		
NPSH AVAILABLE.		0.29		
Source Press	psia	–1.29		
-(Vap Press ↓ suct losses) sub total Psia\Pt	psia	–1.00		
Elev of liquid - pump CL	Pt	45.00		
NPSH Available	Pt	40.40		

SKETCH



DATED 24.3.92 REV 0 DATED ____ REV _____.
NOTES (1) Based on Rated Flow.

Figure 18.19. A centrifugal pump calculation sheet.

on the process flow diagram, while the max temperature is that used for the pump design conditions. It should be the same as the design temperature of the vessel the fluid is pumped from.

Specific gravity @ PT: This is self explanatory. Note the item also calls for the SG @ 60°F.

Vapor pressure @ PT: This is read from the vapor pressure curves given in the appendix in Chapter 3 Appendix Figure 1. First locate the vapor pressure of the stream at atmospheric pressure. (This is the material's normal boiling point). Follow the temperature line down or up to the PT and read off the pressure at that point.

Viscosity @ PT: This too is self explanatory. Note this calculation sheet requires the viscosity to be in *Centipoise*. This is Centistokes × SG.

US gpm @ PT: This is the pump capacity and three rates are asked for. These are:

- *Minimum rate:* The anticipated lowest rate the pump will operate at for any continuous basis. This rate sets the control valve range.
- *Normal rate:* This is the rate given in the material balance and the basis for the hydraulic analysis.
- *Maximum rate:* This is normally set based on the type of service that the pump will undertake. For example: Pumps used only as rundown to storage will have a max rate about 10% above normal. Those used for reflux to towers will have between 15% and 20% above normal.

Discharge pressure, psig: This figure is calculated in the column below. It will also have been determined by the hydraulic analysis of the system.

Suction pressure, psig: Two pressures are asked for in this item. Rated pressure is that calculated in the column below and in the hydraulic analysis given in item 1. It is based on the 'Norm' Rate. The 'Max' suction pressure is based on a source pressure at the *design* pressure rating of the vessel the pump is taking suction from.

Differential pressure, psi: This is the discharge pressure minus the rated suction pressure.

Differential head feet: The head is determined from the differential pressure by the equation:

$$\frac{\text{Diff Press (psi)} \times 144}{62.2 \times \text{SG @ PT}}.$$

NPSH feet: Is calculated in the column below. This is the suction head available greater than the fluid vapor pressure (at the PT) at the pump impeller inlet.

Hydraulic horsepower: This is calculated from the weight per unit time (usually minutes or seconds) of fluid being pumped times the differential head in feet divided by 550 ft-lbs/sec or 33,000 ft-lbs/min. The differential head is always based on the rated suction pressure and the weight on the rated capacity (gpm) for this calculation.

Corrosion/erosion: The process engineer notes any significant characteristic of the fluid regarding its corrosiveness or abrasiveness here.

Turbine conditions

Although this item is not strictly part of a pump definition it should be included for completeness in the case of turbine drives. The data required to complete this item is self explanatory.

Pump material

The process engineer indicates here the acceptable material for the pump in handling the fluid. For example: carbon steel, or cast iron, etc; it is not necessary to specify grade of steel, etc.

The calculation columns

The objective of this section of the calculation sheet is to itemize all the data that are used to provide the figures given in the operating conditions described in Section "Operating conditions for each pump" above. The first column lists those items while the other three columns are available for entering the corresponding numbers. These three columns are provided to cater for alternate conditions that may need to be studied. A space is left on the right of the form to sketch the pumping system (it is very advisable to do a sketch).

The first column starts with the destination pressure, and continues down with the list of the pressure drops in the system to the pump discharge. This section of the column ends with the sum of the pressure drops giving the pump discharge pressure. The items that make up the pump suction pressure are listed next. This starts with he source pressure (usually a vessel) and its static head above the pump. All the pressure drops in the suction side are listed and deducted from the sum of the source pressure and static head to give the pump suction pressure.

The last section in the column itemizes the data that gives the *available* NPSH for the pump. The development of the NPSH is self explanatory.

Centrifugal pump seals

A pump seal is any device around the pump shaft designed to prevent the leakage of liquid out of or air into a pump casing. All industrial pumps have shafts protruding through the casings which require sealing devices. Pump sealing devices are usually either a “packed box” with or without a lantern ring, or a mechanical seal. Controlled leakage is a system sometimes used.

A flushing stream must be introduced into the pump seals for one or more of the following reasons:

- To effect a complete seal
- To provide cooling, washing or lubrication to the seal
- To keep grit from the seal
- To prevent corrosive liquid from reaching the seal

The facilities for accomplishing this are called “flushing system”, and there are two types of these in general use.

- A dead-end system
- A through system

In a “dead-end” system the flushing liquid enters the casing through the stuffing box and combines with the pumped fluid (see Figure 18.20a). A “through” system is one in which the flushing liquid is re circulated between a double seal arrangement and does not enter the pump (see Figure 18.20b). The liquid source may be external to the pump or as on most mechanical seals is a self-flushing system in which the pumped liquid is used as the flushing fluid.

A description of each of the types of sealing devices is presented below and illustrated in Figure 18.21.

Packed boxes (without lantern ring) Figure 18.21a.

This is the simplest type of pump seal. Its principal components are a stuffing box, rings of packing, a throat bushing, and a packing gland. A slight leakage through the packing is required at all times to lubricate the packing. A water quench is used at the packing gland if the packing “leakage” is considered flammable or toxic.

Packed box (with lantern ring) Figure 18.21b

When a packed box pump seal is used in conjunction with a flushing oil system, a lantern ring is usually provided. This metallic ring provides a flow path for the flushing oil to reach the pump shaft. For very erosive or corrosive services, the lantern ring is

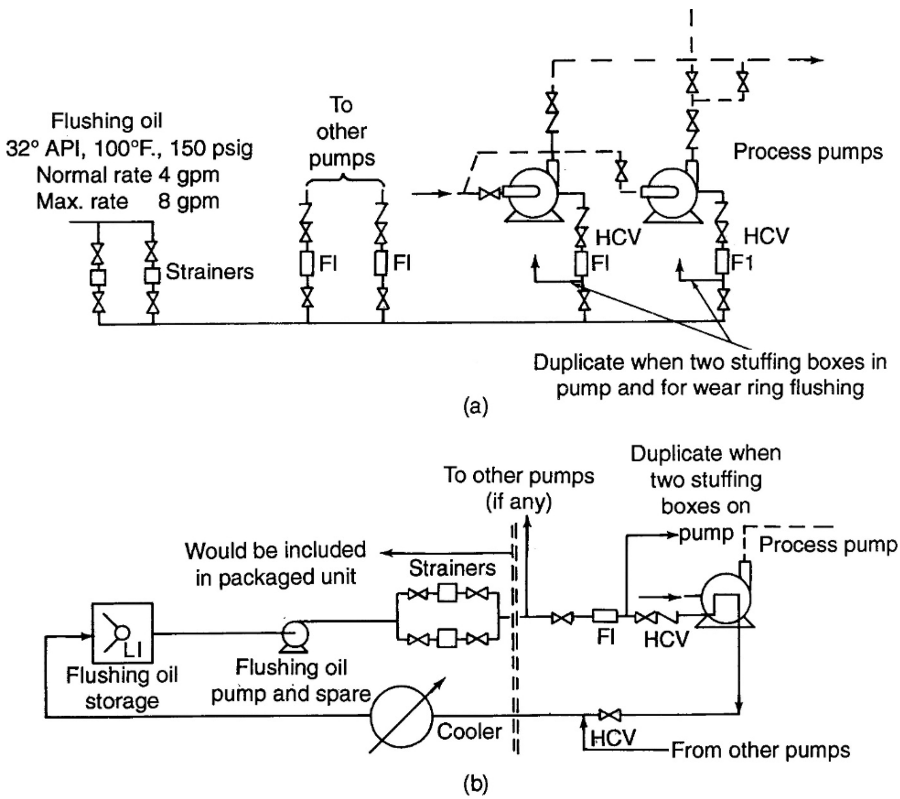


Figure 18.20. Typical flushing systems (a) a dead end system, (b) a through—re-circulating system.

often located next to the throat bushing and a liquid is injected into the throat bushing to prevent the pumped fluid from reaching the packing area. For a pump operation with vacuum suction conditions, the lantern ring is installed at the middle of the box and liquid is injected to prevent air entering the system. This type also operates with positive leakage with the same comments as the packed box without lantern ring.

Mechanical seals Figure 18.21c and d.

Typical basic elements of a single seal are shown in Figure 18.21c. Sealing is affected between the precision-lapped faces of the rotating seal ring and stationary seal ring. The stationary seal ring is usually carbon, and is mounted in the seal plate by an "O" ring. The two "O" ring packing serves the dual purpose of sealing off any liquid tending to leak behind the seal rings, and also to provide flexibility in allowing the seal faces to align themselves exactly so as to compensate for any slight "wobble" of the rotating seal face caused by shaft whip.

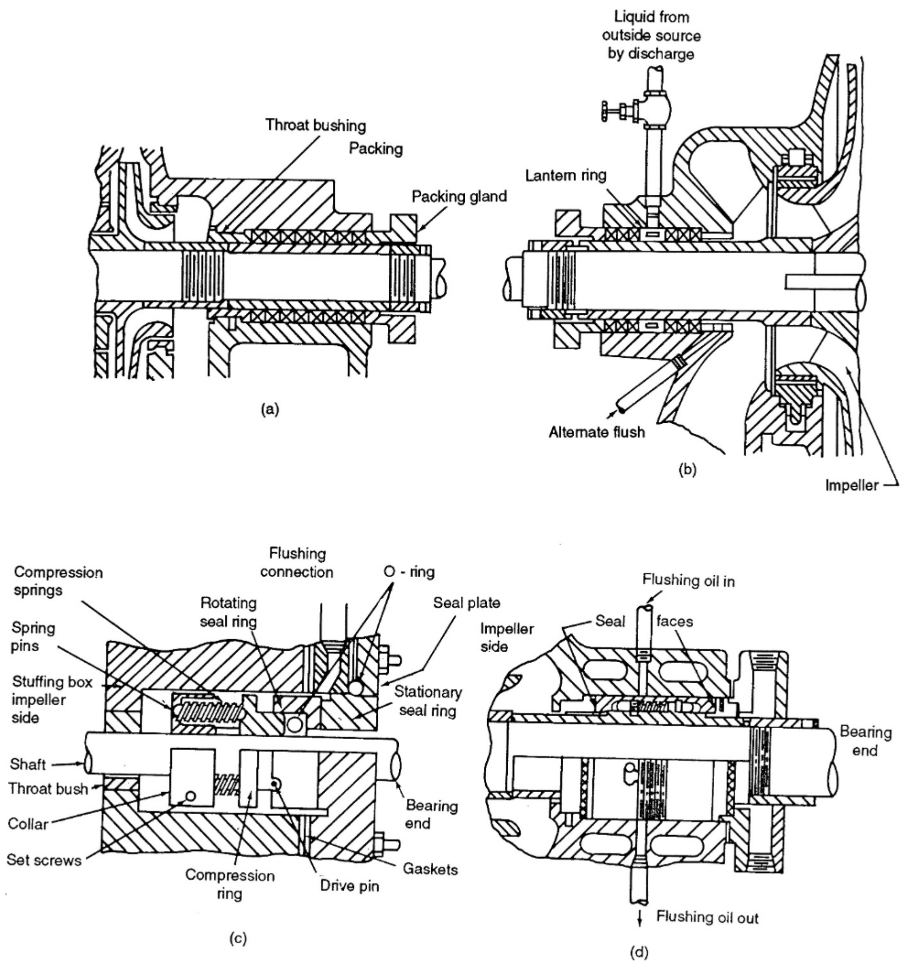


Figure 18.21. Pump shaft packing and seals. (a) Stuffing box completely filled with packing no lantern ring. (b) Externally sealed stuffing box. (c) Single Mechanical Seal. (d) Double Mechanical Seal.

The rotating seal ring is usually stainless steel with a Stellite face. The springs furnish the necessary force to set the “O” ring and hold the seal faces closed under low stuffing box pressures. Any pressure in the box exerts additional force on the rotating sealing ring. The seal is frequently “balanced” so that the face pressure is in correct ratio to the liquid pressure to ensure adequate sealing without excessive loading of the faces. Flushing oil enters the stuffing box through a connection in the seal plate.

A double seal consists of two single seals back to back. See Figure 18.21d. As a double seal is more expensive and requires a complicated seal-oil system, it is used only where single seals are not practical.

Table 18.20 summarizes the application of the various sealing systems in the refinery operation.

Pump drivers and utilities

Most pumps in the process industry are driven either by electric motors, or by steam, usually in the form of steam turbines. This item deals with the calculation of the driver requirements and its specification.

Electric motor drivers

Electric motors are by far the most common pump driver in industry. They are more versatile and are cheaper than a comparable size of steam turbine. The electric motors used for pump drivers are the induction type motor. They range in size from fractional horsepower to 500 and higher horsepower. Sizing the required motor for a pump driver takes into consideration the pump brake horsepower, the energy losses occurring in the coupling device between the pump and the motor, and a contingency factor of about 10%. These are expressed by the equation:

$$\text{Minimum driver BHP} = \frac{\text{Maximum pump BHP} \times 1.1}{\text{Mechanical efficiency of coupling}}$$

If the pump is driven through a direct coupling the efficiency will be 100%. With gears or fluid coupling the efficiency will be between 94% and 97%.

Specifying motor driver requirements

Process engineers are called upon very often to specify pump driver requirements or to check the already existing. In doing this two items of data need to be obtained or calculated. These are:

- The actual required horsepower of the pump motor to drive the pump at its specified duty
- What is actually installed in terms of horsepower

These data are tabulated in terms of power load as follows:

Operating load, kW—This is power input to the motor a normal operating horsepower.
Connected load, kW—This is power input to the motor at motor rated horsepower.

If the pump is spared by another motor driven pump then the connected load will be the sum of *both motors.*

$$\textit{Operating load} = \frac{\textit{Minimum required driver HP} \times .746}{\textit{Efficiency of the motor @ its operating HP}}$$

$$\textit{Connected load} = \frac{\textit{Rated motor HP} \times .746 \times \textit{number of motors}}{\textit{Efficiency of the motors @ 100\% Full load}}.$$

Table 18.20. Application of pump sealing systems

Pumped fluid	Conditions of service	Shaft seal
Clean hydrocarbon or chemical	Suction pressure to 600 psig temperature minus 60°F and lower	Double mech. seal
	Minus 60°F to + 400°F	Single mech. seal self-flushing
	(Solidifies at ambient)	External flush
	400°- 600°F	Single mech. seal 1. Self-flush with cooling 2. External flush
	(Solidifies at ambient)	External flush
	600°- 700°F	Packing
	(Vacuum)	Packing + seal liquid
	700°F and above	Packing
	(Vacuum)	Packing + seal liquid
	Suction pressure to 600 - 1500 psig	Double mech. seal
	Suction pressure above 1500 psig	Special designs
Any dirty or non-lubricating Hydrocarbon or chemical	Pressures to 600 psig pumping temp. minus 60°F and below	Double mech. seal
	Minus 60°- 600°F	1. Single mech. seal with external flush 2. Packing with external flush
	(Flushing liquid not compatible	Double mech. seal
	600°F and above	Packing with external flushing and cooling
Corrosive chemicals without solids	Temperature minus 60°F and below	Double mechanical
	Minus 60°F and 400°F	Single mech. seal
Any slurry	All conditions	Packing with external flush plus wear ring and flushing
Water	To 600 psig Temperature to 160°F	Mechanical seal
	Above 160°F	Mechanical seal with cooling, self or

Table 18.21. Electric motor size and efficiency

Motor rating BHP	Motor connected load, kW	Motor efficiency at a % of full load		
		50	75	100
1	0.98	68	74	76
1.5	1.42	72	76.5	79
2	1.86	73	78	80
3	2.76	77.5	81.5	82
5	4.39	83	82	85
7.5	6.65	81	83.5	84
10	8.78	84	85	85
15	13.0	85	86	86
20	17.05	86.5	87.5	87.5
25	21.0	87.5	88.5	88.5
30	25.1	88	89	89
40	33.5	88	89	89
50	41.7	88	89.5	89.5
75	62.1	89	90	90
100	82.0	84	89	91
125	102.0	85	89.5	91.5
150	123.0	86	89	91
200	161.0	88	91	92.5
250	201.0	90.5	92.5	92.5
300	241.0	90.5	92.5	92.8
350	281.0	90.9	92.6	92.9
400	320.0	91.1	92.8	93.1
450	360.0	91.2	93	93.2

Table 18.21 gives the motor sizes, and standard efficiencies at % of full load. (Higher-efficiency and variable speed motors may also be available.)

Example calculation
Calculate the operating and connected loads for pump 11-P-3 A&B as specified in Figure 18.19 of this chapter.

From the pump calculation sheet the hydraulic horsepower is calculated:

$$\begin{aligned} \text{HHP} &= \frac{\text{lbs/min} \times \text{ diff head in feet}}{33{,}000} \\ &= \frac{8{,}665 \times 264.3}{33{,}000} \\ &= 69.4 \text{ HP} \end{aligned}$$

From Figure 18.19 and assuming 60 cycle pump speed pump efficiency is 79%

$$\begin{aligned} \text{Then brake horsepower} &= \frac{\text{Hydraulic Horsepower}}{.79} \\ &= 87.8 \text{ HP} \end{aligned}$$

This will be a direct driven pump thus coupling efficiency is 100%

$$\text{Minimum motor size} = \text{BHP} \times 1.1 = 96.6 \text{ HP}$$

The closest motor size to this requirement is 100 HP (Table 18.21). This is a little too close so a motor size of 125 HP will be selected.

$$\text{Operating Load} = \frac{\text{Rated HP} \times 0.746}{\text{Efficiency @ \% of full load.}}$$

$$\text{\% of Full load} = \frac{87.8}{125} = 70.2\%$$

$$\text{Efficiency} = 89\% \text{ (from Table 18.21)}$$

$$\begin{aligned}\text{Operating load} &= \frac{87.8 \times .746}{0.89} \\ &= 73.6 \text{ kW}\end{aligned}$$

$$\begin{aligned}\text{Connected load} &= \frac{125 \times .746}{0.915} \\ &= 101.9 \text{ kW say } 102 \text{ kW}\end{aligned}$$

Note if both regular and spare pumps were motor driven then the connected load would be $2 \times 102 = 204$ kW.

Reacceleration requirement

To complete the specification for the motor requirement a degree of process importance of the pump must be established and noted. Voltage drops that can occur in any system may be sufficient to stop the pump. The process engineer must determine how important it is to the process and the safety of the process to be able to restart and reaccelerate the particular pump quickly. The following code of importance has been adopted:

- Re acceleration absolutely necessary.—A
- Re acceleration desirable.—B
- Re acceleration unnecessary.—C

The 'A' category involves any pump critical to keeping the process on stream safely and with no possibility of equipment damage. The 'B' category applies to those pumps that in operation with the 'A' category will maintain the unit 'on spec'. The 'C' category refers to those pumps that can be started manually without any problems.

In the case of the example pump 11-P-3 A&B given here, the service required of the pump is so critical to the operation and orderly shutdown of the process in the case of power failure that the spare pump is Turbine driven. Thus the motor driven regular pump need only be coded 'C' for reacceleration.

Steam turbine drivers

Steam turbines are the second most common pump drivers in modern day process industry. Although more expensive than the electric motor they offer an excellent stand-by to retain the maximum process 'on stream' time. The one big disadvantage with power driven pumps is the reliability of power availability. Steam turbines therefore offer a good alternative in cases of power failure. Another alternative means of pump drivers is the diesel engine or gas engine, but these require their own fuel storage etc. and are certainly not as reliable as the steam turbine.

Most process plants therefore spare the critical pumps in the process with a turbine driven unit which may be started automatically on low process flow.

The principle of the turbine driver

Turbines are the most flexible of prime movers in today's industry. Their horsepower output can be varied by the number and size of the steam nozzles used, speeds can be changed readily, and high speeds without gearing are possible. They have a very wide range of horsepower applications. The operation of the steam turbine is analogous to that of a water wheel where buckets are attached to the wheel which collect the water. The wheel is moved downwards by the weight of the water collected and thus cause the rotation of the wheel. In steam turbines the buckets are replaced by vanes which are impinged by the motive steam to cause the rotating motion. Turbines may consist of one set of vanes keyed to the shaft in the case of a single stage machine or several sets of vanes in the case of multi stage machines. These sets of vanes are called simply 'wheels' and the number of stages are referred to as the number of wheels.

In the case of multistage turbines, the steam leaving the first wheel is directed towards a set of stationary vanes attached to the casing. These stationary vanes reverse the steam flow and serve as nozzles directing the steam toward the second wheel attached to the same shaft.

Most turbines used on a regular basis in a process plant are single stage. Multi stage machines are more efficient but are also much more expensive. Their use therefore is for drivers requiring horsepower in excess of 300. The power industry is a good example for the use of large multi stage turbine drivers. Single or multi stage turbines may be operated either condensing or non condensing. However pump drivers should

not be made *condensing* without a rigorous review to see if other types of drives can be used. The complexity of condensing is hardly worth the small savings in utilities that are made.

The performance of the steam turbine

The salient factors in the performance of the steam turbine are:

- Horsepower output
- Speed
- Steam inlet and outlet conditions
- Its mechanical construction (e.g., number of wheels, size of the wheel, etc.)

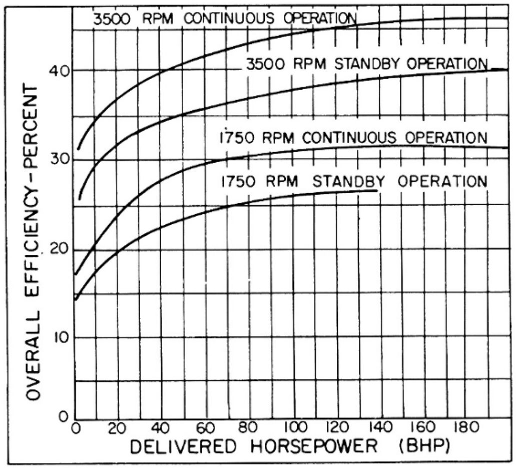
These factors are interrelated and their effect on the performance of the turbine is reflected by a change in over-all efficiency. The *overall efficiency may be defined as the ratio of the energy output to the energy of the steam theoretically available at constant entropy as obtained from a Mollier diagram.* This over-all efficiency is the product of mechanical and thermal efficiencies. The losses in turbines are partly due to friction losses of the rotating shaft and partly to thermodynamic losses and turbulence. Figure 18.22 gives the overall efficiencies plotted against delivered horsepower.

The steam required by a turbine for a given horsepower application is called its "Water Rate". The actual water rate for a turbine is supplied by the manufacturer from test runs carried out on the actual machine in the workshop. Plant operators and other engineers very often need to be able to estimate these water rates for their work. A typical situation arises when determining the best steam balance for a plant. Such estimates may be obtained from Figures 18.23a and b. This and the accompanying notes are self explanatory.

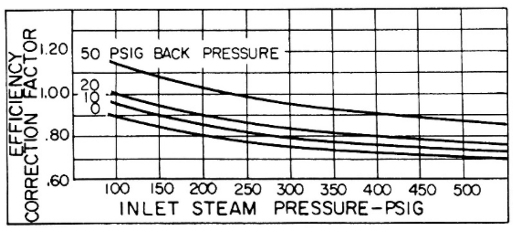
Cooling water requirements

Many pumps in process service require water cooling to various parts of the pump. This cooling water is applied to bearings, stuffing boxes, glands, and pedestals. The application of the cooling water is determined by the manufacturer in accordance with his standard for the service and conditions that the pump must satisfy. Most of the cooling water may be recovered in a closed cooling water system. However gland cooling water is never recovered but is routed to the waste water drain. The following lists the approximate cooling water requirements for pumps and steam turbines:

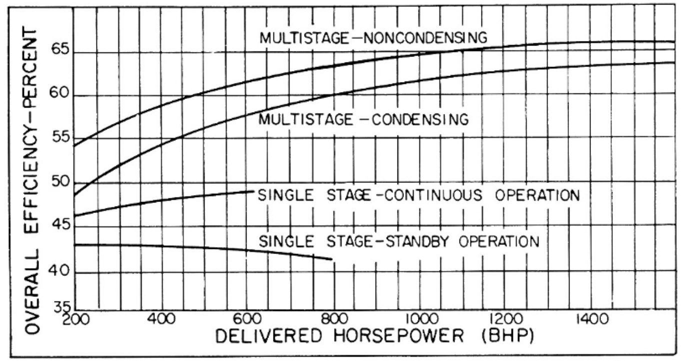
Pumps	*To 1,000 gpm*	*Above 1,000 gpm*
Up to 350 F	0 gpm	0 gpm
350 F–500 F	2 gpm	4 gpm
Above 500 F	3 gpm	6 gpm



GRAPH 1
EFFICIENCY OF TURBINES AT 110 PSIG SAT. STEAM INLET WITH 20 PSIG STEAM EXHAUST



GRAPH 2
STEAM CONDITION CORRECTION FACTOR FOR GRAPH 1



GRAPH 3
EFFICIENCY OF HIGH HORSEPOWER TURBINES

Figure 18.22. Steam turbine efficiencies.

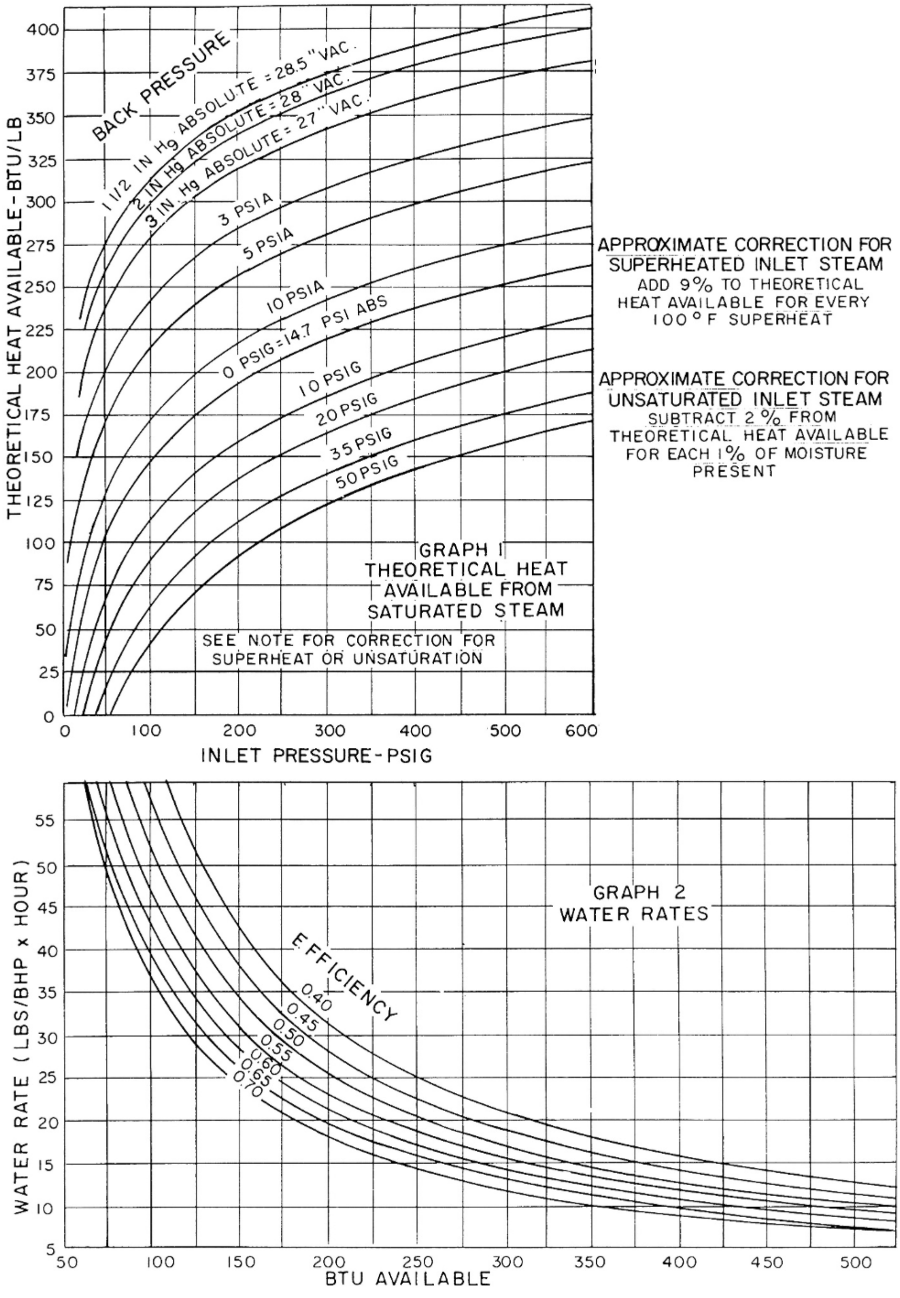


Figure 18.23. Water rates for condensing and non-condensing turbines.

Steam Turbines.

450 F	0 gpm	0 gpm
Above 450 F	3 gpm	3 gpm

18.3 Compressors

Types of compressors and selection

Compressors are divided into four general types, these are:

- Centrifugal
- Axial
- Reciprocating
- Rotary

The name given to each type is descriptive of the means used to compress the gas and comparison of the different types of compressors and typical applications is shown in Table 18.22. A brief description of each of the types now follows:

Centrifugal

This type of compressor consists of an impeller or impellers rotating at high speed within a casing. Flow is continuous and inlet and discharge valves are not required as part of the compression machinery. Block valves are required for isolation during maintenance.

Centrifugal compressors are widely used in the petroleum, gas, and the chemical industries primarily due to the large volumes of gas that frequently have to be handled. Long continuous operating periods without an overhaul make centrifugal compressors desirable for use for petroleum refining and natural gas applications. Normally they are considered for all services where the gas rates are continuous and above 400 ACFM (actual cubic feet per minute) for a clean gas, and 500 ACFM for a dirty gas. These rates are measured at the discharge conditions of the compressor. Dirty gases are considered to be gases similar to those from a catalytic cracker, which may contain some fine particles of solid or liquid material.

The slowly rising head-capacity performance curves make centrifugal compressors easy to control by either suction throttling or variable speed operation.

The main disadvantage of this type compressor is that it is very sensitive to gas density, molecular weight and polytropic compression exponent. A decrease in density or

molecular weight results in an increase in the polytropic head requirement of the compressor to develop the required compression ratio.

Axial flow

These compressors consist of bladed wheels that rotate between bladed stators. Gas flow is parallel to the axis of rotation through the compressor. Axial flow compressors become economically more attractive than centrifugal compressors in applications where the gas rates are above 70 000 ACFM at *suction* conditions. The compressors are extremely small relative to capacity and have a slightly higher efficiency than the centrifugal. Axial flow compressors are widely used as air compressors for jet engines and gas turbines.

Reciprocating

Reciprocating compressors are widely used in the petroleum and chemical industries. They consist of pistons moving in cylinders with inlet and exhaust valves. They are cheaper and more efficient than any other type in the fields in which they are used. Their main advantages are that they are insensitive to gas characteristics and they can handle intermittent loads efficiently. They are made in small capacities and are used in applications where the rates are too small for a centrifugal. Reciprocating compressors are used almost exclusively in services where the discharge pressures are above 5,000 psig.

When compared with centrifugal compressors, the reciprocating compressors require frequent shutdowns for maintenance of valves and other wearing parts. For critical services this requires either a spare compressor or a multiple compressor installation to maintain plant throughput. In addition, they are large and heavy relative to their capacity.

Rotary

Recent developments in the rotary compressor field have opened up areas of application in the process industry with the use of the following types of rotary compressors:

(1) *High pressure screw*
These compressors have been developed into heavy duty type machines. They consist of two rotating helices that rotate in a casing without actual contact. Rotary compressors are lower in cost and have a higher efficiency than centrifugal compressors. They are not sensitive to gas characteristics since they are positive displacement machines. Parts are standardized production items so that a spare rotor is not generally required to be stocked for emergency replacement.

This compressor is noisy and sensitive to temperature rise along the screws due to the close clearances involved. They are good for fouling services where the fouling material forms a soft deposit. This decreases the clearances and leakage along the screws and casing. They are not recommended for use in fouling services in which the deposits are hard.

Variation in speed or a discharge bypass to suction are the only types of control that can be used.

(2) *Low pressure screw, lobe and sliding vane*

These compressors should be used only for low pressure, light duty, non-critical applications. They operate on the same principle as the high pressure screw type but have different mechanical designs. The same advantages and disadvantages apply as those for rotary high pressure screw compressors. They are even lower in cost than the high pressure screw compressors but contain parts having limited life, thus requiring more maintenance.

Only centrifugal and reciprocating compressors will be discussed further in this book.

Calculating horsepower of centrifugal compressors

Centrifugal compressors are used in process service where high capacity flows are required. A typical example is the recycle compressor for handling a hydrogen rich stream in some oil refining and petrochemical processes. The following table gives some idea of the centrifugal compressors capacity range.

Centrifugal compressor flow range

Nominal flow range (inlet acfm)	Average polytropic efficiency	Average adiabatic efficiency	Speed to develop 10,000 ft head/wheel
500–7,500	0.74	0.70	10,500
7,500–20,000	0.77	0.73	8,200
20,000–33,000	0.77	0.73	6,500
33,000–55,000	0.77	0.73	4,900
55,000–80,000	0.77	0.73	4,300
80,000–115,000	0.77	0.73	3,600
115,000–145,000	0.77	0.73	2,800
145,000–200,000	0.77	0.73	2,500

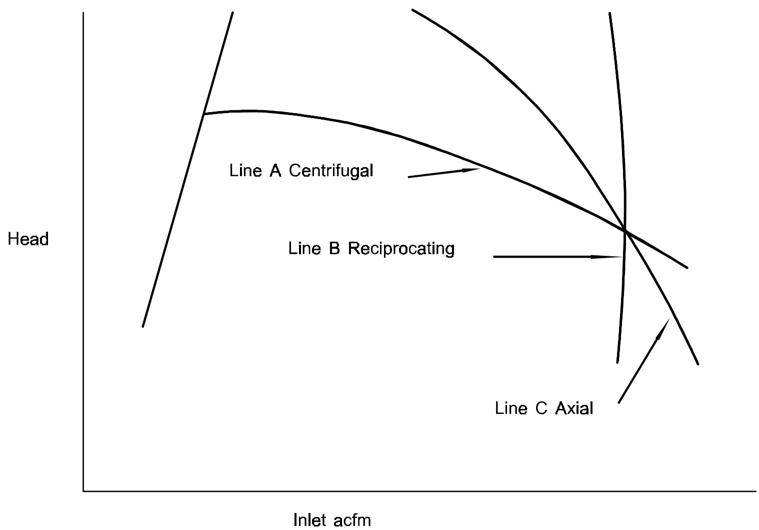
In general the head or differential pressure levels served by centrifugal compressor is considerably lower than that for reciprocal. The following diagram illustrates this feature.

Table 18.22. Comparison of compressors and typical applications

Type and control lable range	Percent availability	Operating speed volumetric capacity compression ratio per stage (6)	Compression eff.	Advantages	Disadvantages	Usual drivers	Common applications
Centrifugal 70–100%	99.5 to 100% (1)	3,000–15,000 RPM (2) 400–500 ACFM minimum @ discharge 150,000 ACFM max. Suction volume (5) 80–100,000 ft. polytrophic head/casing	70–78%	1. Long continuous operating periods 2. Low maintenance costs 3. Small size relative to capacity 4. Ease of capacity control	1. Pressure ratio is sensitive to gas density and molecular weight 2. Spare rotor required	Steam turbine Gas turbine Electric motor Waste gas Expander	Large refrigeration system Cat cracker air Large catalytic reformer recycle gas
Axial flow 80–100%	99.5 to 100%	4,000–12,000 RPM (2) 70,000 min. ACFM 2–4 compression ratio per casing	75–82%	1. Very high throughputs possible 2. Extremely small size relative to capacity 3. Higher efficiency than centrifugals 4. Good for parallel operation with other axials or centrifugals	1. Capacity flexibility limited by steep head-capacity curve and short stable operating range except when variable pitch stators are used 2. Performance and efficiency are sensitive to fouling 3. Spare rotor and spare stator blading are required	Steam turbine Gas turbine Electric motor Waste gas Expander	Cat cracker air (large)
Reciprocating See Sect. D for Range	98% clean gas (3) 95% dirty gas (3) 95% clean gas (4) 93% dirty gas (4)	300–1,000 RPM 5 max. compression ratio or 330–380°F max. discharge temp.	75–85%	1. Handles intermittent loads efficiently 2. Lower cost for small capacities 3. Used for very high discharge pressures (up to 50,000 psig) 4. Higher efficiency than centrifugal in lower capacity ranges 5. Insensitive to gas characteristics	1. Short continuous operating periods require spare or multiple machine installations if service is critical 2. Higher maintenance costs than centrifugal 3. Pulsation and vibration require engineered piping arrangement 4. Availability decreases when non-lubricated machines required to avoid lubricating oil in gas discharge	Synchronous motor Coupled or integral electric motor Coupled or integral engine	Instrument air Refinery air Fuel gas Synthesis gas Crude gas Small catalytic reformer recycle gas. Small refrigeration system Low Mole. Wt. gas

Rotary High Pressure Screw 55–100%	99–99.5	2,500–10,000 RPM 1,000–20,000 ACFM @ suction 4 to 7 max. compression ratio per casing but not exceeding 100 psi ΔP	75–80%	1. Lower cost than centrifugals 2. Higher efficiency than centrifugals 3. Not sensitive to gas characteristics 4. Parts are standardised production items and no spare rotor required	1. Noisy, require inlet and discharge silencers 2. Sensitive to temperature rise due to close clearances 3. Not recommended for use where fouling produces hard deposits 4. Speed or bypass control are only type applicable	Electric motor Steam turbine Gas turbine Waste gas expander	Refinery air Fuel gas Cat. Cracker Air (small)
Low Pressure Screw, Lobe and Vane type Rotaries Fixed Capacity	Not recommended for continuous service	1,500–3,600 RPM 100–12,000 ACFM suction 2 compression ratio per stage 50 psi max. discharge pressure	75–80%	1. Low first cost 2. Low maintenance cost 3. Parts are standardised production items and no spare rotor is required	1. Limited life 2. Speed or bypass control are only type applicable 3. Very noisy	Electric motor Steam turbine	Low pressure, light duty, non-critical services

Note: Clean service machines have the highest availability.
Large machines run at lower speeds.
Between turnarounds of 3 days every 8–12,000 hr with electric drive. 95–98% includes 8 hr shutdowns every few months for valve maintenance.
Between turnarounds of 2 weeks every 8–12,000 hr with engine drive. 93–95% includes 8 hr shutdowns every month for maintenance checks on compressor valves and engine driver.
Axial flow compressors should be considered at gas rate above 70,000 ACFM @ suction conditions.
Stages can be compounded in series for higher rates.



Process Engineers are often required to establish the capability of a centrifugal compressor in a particular service or to assess the machine's capability to handle a different service. In conducting these studies it is necessary to determine the machine's horsepower under the study conditions. This item provides a procedure where the gas horsepower (and thereafter the brake horsepower) of a compressor can be calculated.

This procedure is as follows:

Step 1. Establish the duty required from the compressor in terms of:

- Capacity in CF/min at inlet conditions
- Design inlet temp
- Design inlet pressure
- The mole wt of the gas to be handled
- Compression ratio (P2/P1) P2 being the discharge pressure and P1 the inlet pressure

Step 2. Establish the 'K' value for the gas. If this is a pure gas (such as Oxygen) the 'K' value can be read from data books. Otherwise the 'K' value is the ratio CP/CV. See Figure 18.A.3 in the Appendix. (*Note*: Do not confuse this 'K' factor with Equilibrium Constants.)

Step 3. Calculate volume of the gas in SCF/min. This is the inlet CFM times inlet pressure times 520 divided by 14.7 psia times inlet temperature in °R, thus

$$\text{SCFM} = \frac{1\ \text{CFM} \times \text{inlet press} \times 520}{14.7 \times \text{inlet temp}\ ^\circ\text{R}}$$

Step 4. Calculate number of moles gas/min by dividing SCF/m by 378. Multiply number of moles by mole wt for lbs/min of gas.

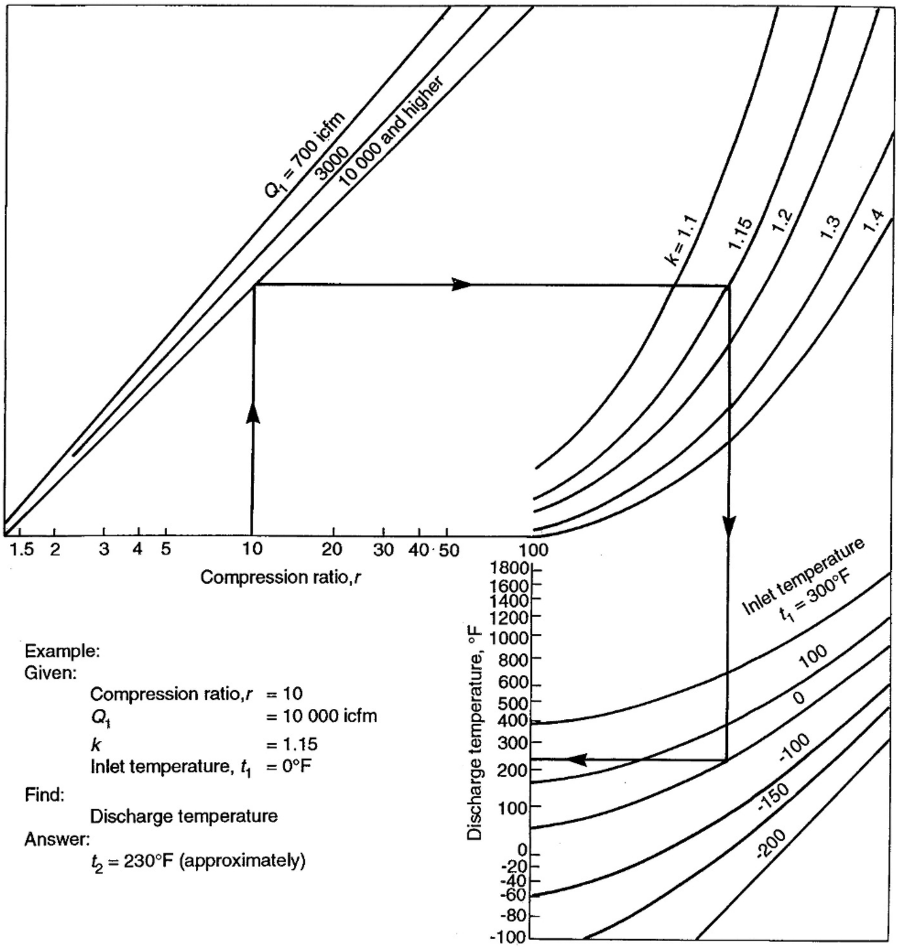


Figure 18.24. Estimated discharge temperatures for centrifugal compressors.

Step 5. Read off the estimated discharge temperature from Figure 18.24 using this and the discharge pressure calculate the volume in CF/min at discharge.

Step 6. Calculate density of gas at suction and discharge using the weight calculated in Step 4 and the CFM for suction and the CF/min calculated in Step 5 for discharge. This density will be in lbs/cuft.

Step 7. The average value for Z is taken as Z at suction + Z at discharge divided by 2. Z (compressibility factor) is calculated by the expression.

where

$$Z = \frac{\mathrm{MP}}{T\rho_{\mathrm{v}} \times 10.73}$$

M = mole weight
P = pressure @ psia
T = °R (°F + 460)
ρ_V = density in lbs/cuft

Step 8. Calculate the adiabatic head in ft lbs/lb using the expression

$$H_{ad} = \frac{Z_{ave} \times R \times T_i}{(K-1)/K}\left[\left(\frac{P_2}{P_1}\right)^{(K-1)/K} - 1\right]$$

where

H_{ad} = the adiabatic head in ft lbs/lb
Z_{ave} = average compressibility factor
R = gas constant = 1,545/mole wt
K = adiabatic exponent CP/CV
P_2 = discharge pressure psia
P_1 = suction pressure psia
T = inlet temperature °R

Step 9. The gas HP is obtained using the expression

$$\text{HP} = \frac{W \times H_{ad}}{\eta_{ad} \times 33{,}000}$$

where

W = weight in lbs/min of gas
H_{ad} = adiabatic head in ft lbs/lb
η = adiabatic efficiency (0.7–0.75)

Step 10. Check GHP using Figure 18.25.

A example calculation now follows:

Example calculation
To determine the Gas HP of a centrifugal compressor assuming isentropic compression:

Compression ratio = 10.0
Capacity (actual inlet CF/min) = 10,000
K_{ave} = 1.15
T_1 °F = 100
P_1 psia = 100
Mole wt = 30
lbs/min = 5,013

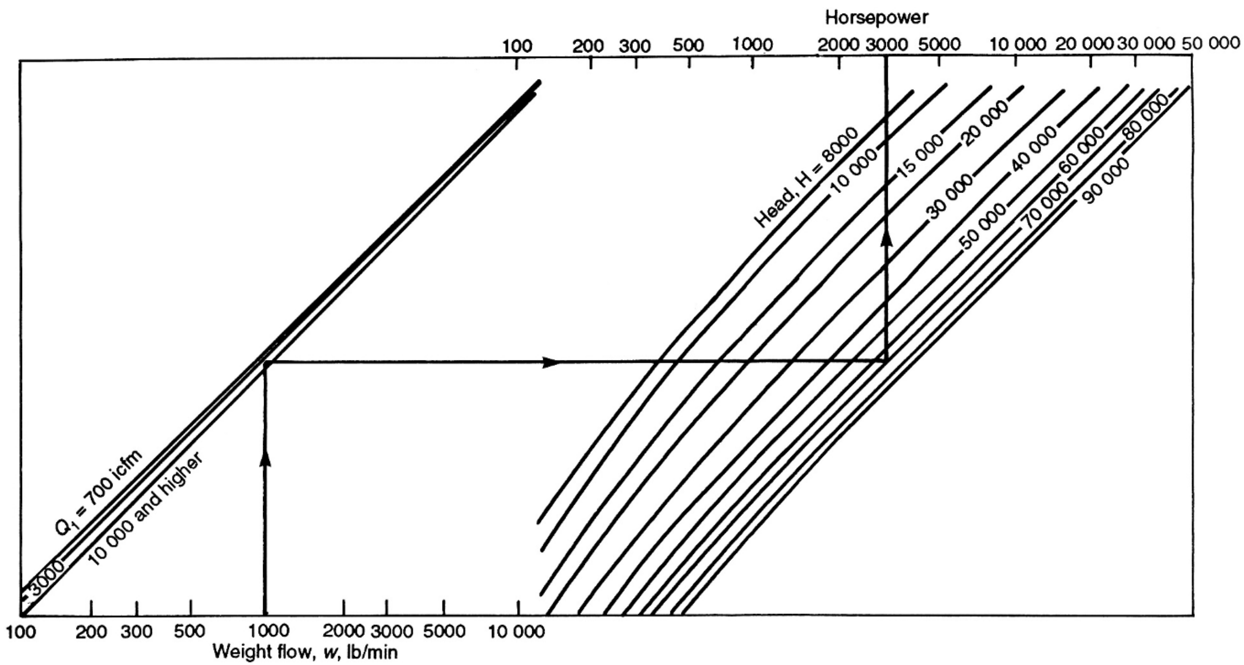


Figure 18.25. Determination of centrifugal compressor horsepower.

For isentropic compression

$$H_{ad} = \frac{Z_{ave} \times R \times T_i}{(K-1)/K} \left[\left(\frac{P_2}{P_1} \right)^{(K-1)/K} - 1 \right]$$

where

H_{ad} = adiabatic head in ft lbs/lb
Z = compressibility factor (ave)
R = gas constant = 1,545/MW
K = adiabatic exponent $CP/CV = 1.15$
T = temperature in °R = °F + 460°F
P_2 = discharge pressure psia
P_1 = suction pressure psia

Z at inlet conditions

$$\rho_V = 0.5 \text{ lbs/cuft}$$

$$Z = \frac{\text{MP}}{T\rho_v} \times 10.73 = \frac{30 \times 100}{560 \times 0.5} \times 10.73 = 0.998$$

Estimated discharge temp (Figure 18.24) = 400°F

$$Z_{dis} = \frac{30 \times 1{,}000}{860 \times 3.26} = 0.997$$

Use 0.998

$$H_{ad} = \frac{0.998 \times 51.5 \times 560}{\frac{0.15}{1.15}} \left[\left(\frac{1{,}000}{100} \right)^{0.13} - 1 \right]$$

$$= 77{,}004 \text{ ft lbs/lb}$$

$$= 220{,}664\ (1.349 - 1)$$

$$\text{Gas HP} = \frac{W \times H_{ad}}{\eta_{ad} \times 33{,}000}$$

Let η_{ad} be 0.75

$$\frac{5{,}013 \times 77{,}004}{0.75 \times 33{,}000} = 15{,}598 \text{ ghp}$$

This compares well with the estimate based on Figure 18.25.

Centrifugal compressor surge control, performance curves and seals

Centrifugal compressors can be counted on for uninterrupted run lengths of between 18 and 36 months after the initial shakedown run. The 18 month run corresponds to a

compressor handling dirty gas, such as furnace gas, and the 36 month run corresponds to a clean gas service, such as refrigerant.

Spare compressors are not usually provided. A spare rotor, however, is required to be stocked as insurance against an extended downtime. Since this rotor is part of the capital cost of the equipment, it is not accounted for as spare parts. Only reliable drivers such as an electric motor, steam or gas turbine can be used where long continuous run lengths are required. In the case of steam and gas turbines, the drivers will probably dictate the maximum possible run length. The high operating speed of a centrifugal compressor also favors the selection of these type of high-speed drivers. The speed of these drivers can be specified to be the same as those of the compressor. For electric motor drives, a speed increasing gear is normally required. Centrifugal compressors can be broadly classified with regards to head and capacity as follows:

	Speed, RPM	Suction, ACFM	Polytropic head ft #/#
Small standard multistage	3,000–3,600	100–1,000	to 8,500
Standard single stage	3,000–3,600	700–60,000	1,000–6,700
Special single stage	3,000–15,000	1,000–60,000	6,700–11,500
Special multistage casing, uncooled	3,000–15,000	1,000–140,000	6,700–100,000
Special multistage, multi-casing, inter-cooled	3,000–15,000	2,500–140,000	37,000 up

As a guide, the maximum head per impeller is about 10,000 ft. Normally, about 8 impellers can be used in a casing.

The minimum allowable volume of gas at the compressor discharge is about 400 ACFM for a clean gas and 500 ACFM for a dirty gas. Dirty gases are considered to be similar to the gas from a steam or catalytic cracking unit.

The discharge temperature is limited to about 250°F for gases that may polymerize and 400°F for other gases. Normally inter-coolers will be used to keep the discharge temperature within these limits. These temperature limitations do not apply to special centrifugal flue gas re-circulator which can be obtained to operate at over 800°F. There is also a temperature rise limitation of 350°F per casing. This is the maximum temperature rise that can be tolerated due to thermal expansion considerations.

Use of cast iron as a casing material is limited to 450°F maximum. Temperatures of −150°F to −175°F can be tolerated in conventional designs. Lower temperatures are not common and will require consulting on individual design features.

Surge

A characteristic peculiar to centrifugal and axial compressors is a minimum capacity at which the compressor operation is stable. This minimum capacity is referred to as

the surge or pumping point. At surge, the compressor does not meet the pressure of the system into which it is discharging. This causes a cycle of flow reversal as the compressor alternately delivers gas and the system returns it.

The surge point of a compressor is nearly independent of its speed. It depends largely on the number of wheels or impellers in series in each stage of compression. Reasonable reductions in capacity to specify for a compressor are shown below.

Wheels/compression stage	% of normal capacity at surge—maximum
1	55
2	65
3 or greater	70

An automatic re-circulation bypass is required on most compressors to maintain the minimum flow rates shown. These are required during start-up or when the normal load falls below the surge point. Cooling is required in the recycle circuit if the discharge gas is returned to the compressor suction.

Performance curves

The rise of performance curves should be specified for a compressor. This is normally done by specifying the pressure ratio rise to surge required in each stage of compression. A continuously rising curve from normal flow rate to surge flow is required for stable control.

The pressure ratio rise to surge is largely a function of the number of impellers per compression stage. Reasonable pressure ratio rises to specify are shown below:

Wheels/compression stage	Minimum % of rise in pressure ratio from normal to surge flow
1	$3\frac{1}{2}$
2	6–7
2 or greater	$7\frac{1}{2}$

Frequently, the performance curves for a compressor have to be plotted to determine if all anticipated process operations will fit the compressor and its specific speed control. Three points on the head-capacity curve are always known. These are the normal, surge and maximum capacity points. The normal capacity is always considered to be on the 100% speed curve of the compressor. The surge point and the compression ratio

rise to surge have been specified. From this the head produced by the compressor at the surge point can be back-calculated using the head-pressure ratio relationship. The maximum capacity point is specified to be at least 115% capacity at 85% of normal head.

The head-capacity curve retains its characteristic shape with changes in speed. Curves at other speeds can be obtained from the three known points on the 100% speed curve by using the following relationships:

1. The polytropic head varies directly as the speed squared.
2. The capacity varies directly as speed.
3. The efficiency remains constant.

Figure 18.26 shows a typical centrifugal compressor performance curve.

Control

Speed
Speed control is the most efficient type of control from an energy consideration. It requires, however, that a variable speed driver such as a steam turbine or gas turbine, or a variable-speed electric motor be used. The compressor is controlled by shifting its performance curve to match the systems requirement.

Suction throttling

- *Adjustable inlet guide vanes.* Adjustable inlet guide vanes are the most efficient method of adjusting the capacity of a constant speed compressor to match the system characteristics. They consist of a venetian blind device that is positioned by a rack and pinion linkage. While the guide vanes do some throttling, their main effect is to change the velocity of the gas to that of the impeller vane by changing the direction of flow. This changes the head produced and in effect changes the characteristic of the machine.
- *Suction throttle.* This control consists of a control valve located in the compressor suction which regulates the suction pressure to the compressor. The control valve results in a greater power loss compared to adjustable inlet guide vane control since it is a pure throttling effect. Suction throttle valves are lower in cost than adjustable inlet guide vanes.
- *Discharge throttling.* This control consists of a control valve located in the compressor discharge. Discharge throttle valves are seldom used since they offer relatively little power reduction at reduced capacity. The effect is simply to “push” the compressor back on its curve.

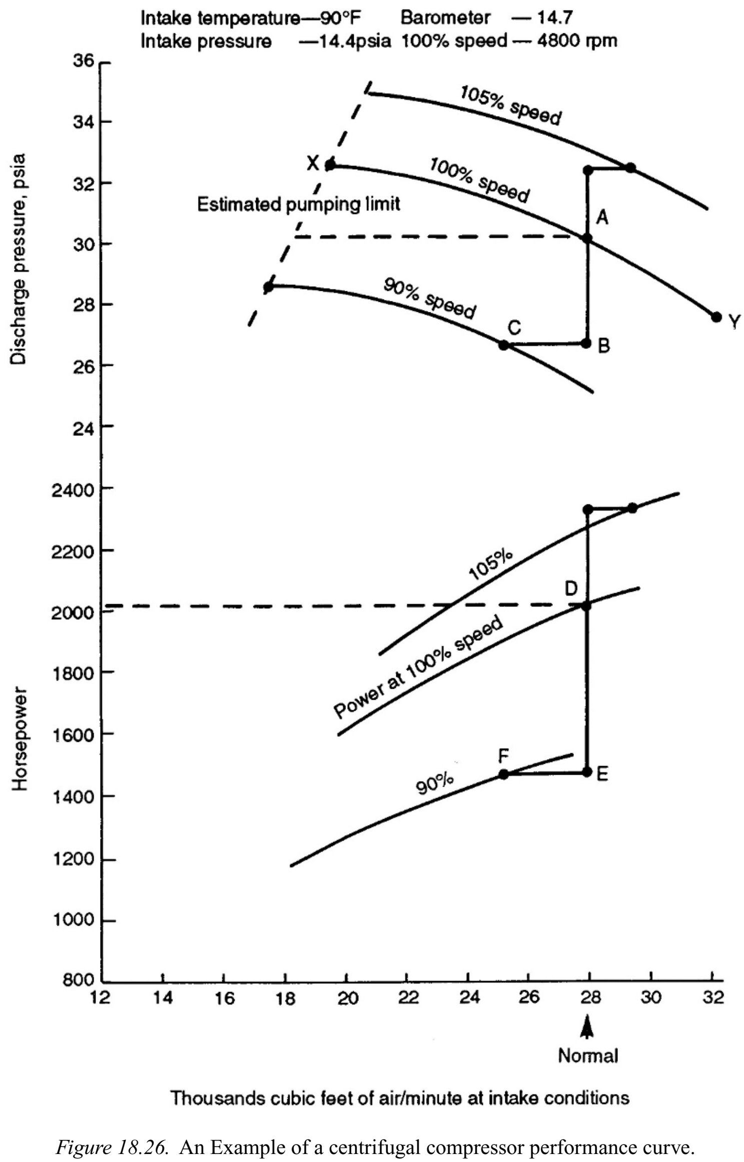


Figure 18.26. An Example of a centrifugal compressor performance curve.

Seals

Table 18.23 shows the types of seals that are commonly used in centrifugal compressors. The start-up as well as the operating conditions of the compressor should be considered in selecting a seal. Often the system is evacuated when hydrocarbons are handled prior to its start-up. This requires that the seal be good for vacuum conditions.

Table 18.23. Centrifugal compressor seals.

No	Application	Gas being handled	Inlet pressure, psia	Seal arrangement
1	Air compressor	Atmospheric air	Any	Labyrinth
2	Gas compressor	Non-corrosive Non-hazardous Non-fouling inexpensive	Any	Labyrinth
3	Gas compressor (note 1)	Non-corrosive or Corrosive Non-hazardous or Hazardous Non-fouling or Fouling	10–25	Labyrinth with injection or ejection of fluid being handled.
4	Gas compressor	Non-corrosive Non-hazardous Non-fouling	All pressures	Oil seal combined with Lube oil system.
5	Gas compressor	Corrosive Non-hazardous or Hazardous	All pressures	Oil seal with seal oil Separate from lube oil System.

Note 1: Where some gas loss or air induction is tolerable.

Specifying a centrifugal compressor

The process specification must give all the information concerning the gas that is to be handled, its inlet and outlet conditions, the utilities that are available and the service that is required of the compressor. The process specification sheet given here shows the minimum that a process engineer should provide in approaching manufacturers. An explanation of this specification now follows covering each of the items in the specification.

Title block

This requires the item to be identified by item number and its title. The number of units that the specification refers to is also given here. For a centrifugal compressor this will normally be just one as very seldom is a spare machine required.

Normal and rated columns

More often than not the conditions and quantities required to be handled will vary during the operation of the machine. The two columns therefore will be completed showing the average normal data in the first column and the most severe conditions and duty required by the compressor in the second. The severe conditions in column two are for a continuous length of operation not instantaneous peaks (or troughs) that may be encountered.

Gas

The composition and gas stream identification must be included as part of the process specification. Usually the composition of the gas is listed on a separate sheet as shown in the example. Note in many catalytic processes that utilize a recycle gas the composition of the gas will change as the catalyst in the process ages. Thus it will be necessary to list the gas composition at the start of the run (SOR) and at the end of the run (EOR).

The compressor may also be required to handle an entirely different gas stream at some time or other. This too must be noted. For example in many petroleum refining processes a recycle compressor normally handling a light predominately hydrogen gas is also used for handling air or nitrogen during catalyst regeneration, purging, and start-up.

Volume flow

This is the quantity of gas to be handled stated at 14.7 psia and 60 F.

Weight flow

This is the weight of gas to be handled in either lbs/min or lbs/hr.

Inlet conditions

Pressure: This is the pressure of the gas at the inlet flange of the compressor in psia.
Temperature: This is the temperature of the gas at the inlet flange of the compressor.
Mole weight: The mole weight of the gas is calculated from the gas composition given as part of the specification.
C_p/C_v: This is the ratio of specific heats of the gas again obtained from the gas mole wt and Table 18.A.1 in the Appendix.
Compressibility factor (z): use the value at inlet conditions calculated as shown in step 7 of the chapter 'Calculating the Horsepower of Centrifugal Compressors'.
Inlet volume: This is the actual volume of gas at the conditions of temperature and pressure existing at the compressor inlet. Thus:

$$\text{ACFM} = \frac{\text{SCFM} \times 14.7 \times (\text{inlet temp F} + 460)}{(60\text{ F} + 460) \times \text{inlet press psia}}$$

Discharge conditions

Pressure: This is the pressure at the compressor outlet flange and is quoted in either psia or psig.
Temperature: This is estimated using Figure 18.24.

C_p/C_v: This will be the same as inlet.
Adiabatic efficiency: Will be as given in Figure 18.27.

Approximate driver horsepower

This item will include the adiabatic (or gas) horsepower as calculated in the section on calculating the horsepower of centrifugal compressors in this chapter plus the following losses:

Leakage loss—1% of Adiabatic HP
Seal losses—35 HP for all HP ranges.
Bearing loss—5 HP for all HP ranges.

The reminder of the spec sheet contains all the essential data and requirements that may affect the duty and performance of the compressor. Much of this is self explanatory however there are some items that require comment. These are:

1. Most compressor installations today are under an open sided shelter with a small overhead gantry crane assembly for maintenance.
2. Usually the lube and seal oil assemblies have their own pump and control systems. Consequently even if the compressor itself is to be steam driven there may still be need to give details of utilities for the ancillary equipment.
3. Details of the gas composition is essential for any development of the compressor. This is listed on the last page of the specification together with any notes of importance concerning the machine and its operation.

An example calculation for a specification sheet follows:

Example calculation
Prepare a process specification sheet for a compressor to handle the hydrogen recycle stream in an o-xylene isomerization plant. Details are as follows:

Fresh feed rate : 5,000 bpsd of m-xylene.
Recycle gas rate : 7,000 Scf of hydrogen per Bbl of Fresh feed.
Gas composition mole % :

	Start of Run (SOR)	End of Run (EOR)
H_2	85.00	68.78
C_1	4.4	9.17
C_2	4.2	8.86
C_3	3.6	7.36
iC_4	0.78	1.62
nC_4	0.99	2.05
C_5s	1.03	2.16

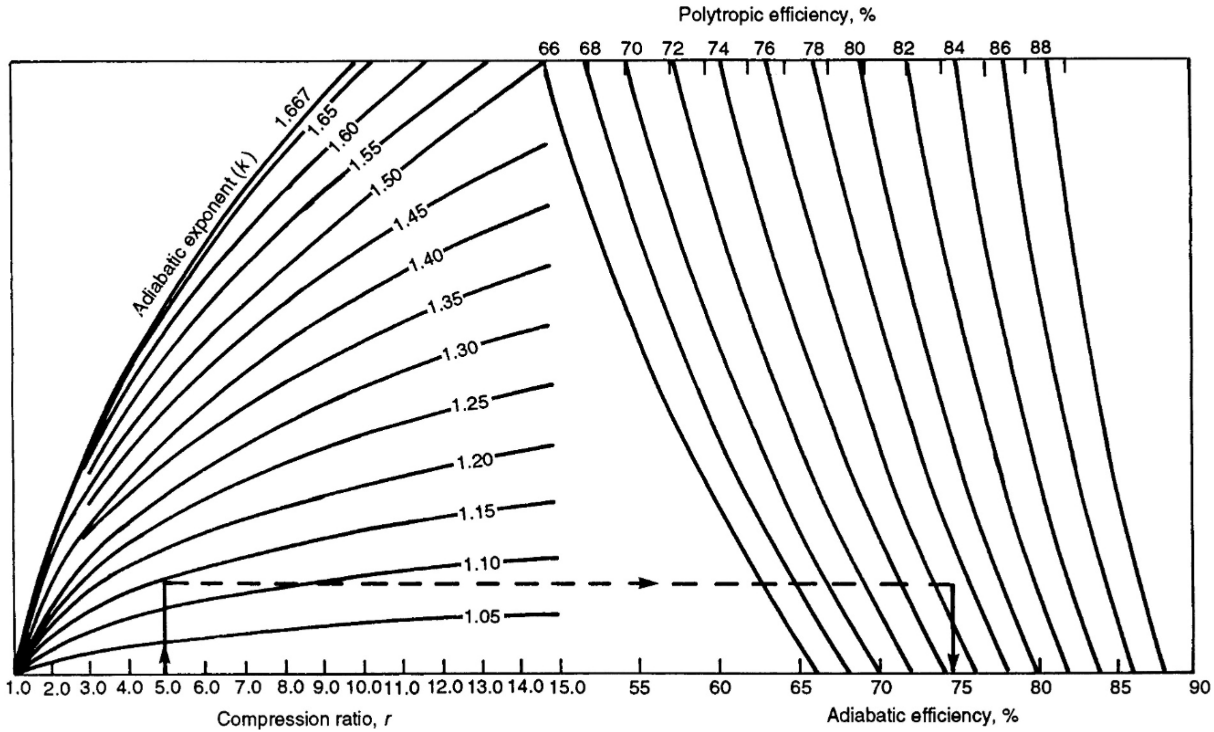


Figure 18.27. Adiabatic efficiencies for centrifugal compressors.

Suction pressure	:	150 psig
Reactor pressure	:	500 psig.
Suction temperature	:	100 F

Step 1. Calculate the mole weight of the gas.

	SOR			EOR	
	mole%	MW	wt Factor	mole%	wt Factor
H_2	85.0	2	170	68.78	138
C_1	4.4	16	70	9.17	147
C_2	4.2	30	126	8.86	266
C_3	3.6	44	158	7.36	324
iC_4	0.78	58	45	1.62	94
nC_4	0.99	58	57	2.05	119
C_5s	1.03	72	74	2.16	156
Total	100.00	700	100.00	1,244	
MW	7.0			12.44	

Step 2. Calculate volume flow of gas in SCF/min.

$$\begin{aligned}
\text{Total volume of } H_2 \text{ required} &= 5{,}000\,\text{BPSD} \times 7{,}000\,\text{SCF} \\
&= 35.00\,\text{mmScf/day} \\
&= 24{,}306\,\text{Scf/min.} \\
\text{For SOR volume gas Flow} &= \frac{24{,}306}{0.85} \\
&= 28{,}595\,\text{Scf/min.} \\
\text{For EOR volume gas flow} &= \frac{24{,}306}{0.6878} \\
&= 35{,}339\,\text{Scf/min.}
\end{aligned}$$

Step 3. Calculate weight flow in lbs/min.

$$\begin{aligned}
\text{moles/min of gas} &= \frac{\text{Scf/min}}{378} \\
\text{For SOR moles/min} &= 75.6 \\
\text{For EOR moles/min} &= 93.5 \\
\text{lbs/min for SOR} &= 75.6 \times 7.0 = 529\ \text{lbs/min} \\
\text{lbs/min for EOR} &= 93.5 \times 12.44 = 1{,}163\ \text{lbs/min}
\end{aligned}$$

Step 4. Calculate ACFM at inlet conditions.

$$\begin{aligned}
\text{Compressor inlet pressure} &= 165\,\text{psia} \\
\text{'' \quad '' \quad Temp} &= 100\,\text{F}
\end{aligned}$$

$$\text{For SOR ACFM} = \frac{28{,}597 \times 14.7 \times 560}{520 \times 165}$$
$$= 2{,}744 \text{ cft/min}$$
$$\text{For EOR ACFM} = \frac{35{,}339 \times 14.7 \times 560}{520 \times 165}$$
$$= 3{,}391 \text{ cft/min}$$

Step 5. Estimate the C_p/C_v ratio.

The molal proportions will be used for this purpose. The ratio for each component will be taken from Table 18.A.1 in the Appendix.

		SOR		EOR	
	C_p/C_v	mole%	C_p/C_v fact	mole%	C_p/C_v fact
H_2	1.40	85.0	119	68.78	96.29
C_1	1.30	4.4	5.7	9.17	11.92
C_2	1.22	4.2	5.12	8.86	10.81
C_3	1.14	3.6	4.10	7.36	8.39
iC_4	1.11	0.78	0.87	1.62	1.80
nC_4	1.11	0.99	1.10	2.05	2.28
C_5S	1.09	1.03	1.12	2.16	2.35
Total		100.00	131.9	100.00	133.84

Then C_p/C_v for the gas is:

$$\text{SOR} = 1.319$$
$$\text{EOR} = 1.338$$

Step 6. Calculate compressibility factors.

$$Z = \frac{MW \times P_1}{T \times \rho_v \times 10.73}$$

$$\text{For SOR flows} \quad Z = \frac{7.0 \times 165}{560 \times 0.193 \times 10.73}$$
$$= 0.996$$
$$\rho_v = \frac{\text{wt lbs/min}}{\text{ACFM}}$$
$$\text{For EOR flows} \quad Z = \frac{12.46 \times 165}{560 \times 0.343 \times 10.73}$$
$$= 0.998$$

Step 7. Calculate outlet temperature.

Approx discharge temperature is read from Figure 18.24 in this chapter using the following:

$$\begin{aligned} \text{ACFM for SOR} &= 2{,}744 \\ \text{ACFM for EOR} &= 3{,}391 \\ \text{Compression ratio} &= \frac{515}{165} \\ &= 3.12 \\ \text{Inlet temp F} &= 100 \end{aligned}$$

Then

$$\begin{aligned} \text{Discharge temp for SOR} &= 370\text{ F} \\ \text{'' \quad '' \quad '' \quad EOR} &= 340\text{ F} \end{aligned}$$

Step 8. Calculate approx driver HP.

$$H_{\text{ad}} = \frac{Z_{\text{ave}} \times R \times T_{\text{i}}}{(K-1)/K} \left[\left(\frac{P_2}{P_1} \right)^{(K-1)/K} - 1 \right]$$

where

H_{ad} = adiabatic head in ft lbs/lb
Z = compressibility factor (ave)
R = gas constant = 1,545/MW
K = adiabatic exponent $C_P/C_V = 1.15$
T = temperature in °R = °F + 460°F
P_2 = discharge pressure psia
P_1 = suction pressure psi

Then for SOR:

Had for SOR conditions = 161,063
Had for EOR conditions = 91,546

Step 9. The gas HP is obtained using the expression

$$\text{Gas HP} = \frac{W \times H_{\text{ad}}}{\eta_{\text{ad}} \times 33{,}000}$$

where

W = weight in lbs/min of gas
H_{ad} = adiabatic head in ft lbs/lb
η = adiabatic efficiency (0.7–0.75)

Let η_{ad} be 0.73

For SOR Gas HP = 3,536
For EOR Gas HP = 4,420

Step 10. The driver HP is as follows:

	SOR	EOR
Gas HP	3,536	4,420
Leakage losses	35	44
Bearing losses	35	35
Seal losses	35	35
DRIVER HP	3,641	4,534

Calculating reciprocating compressor horsepower

Reciprocating compressors are used extensively in the process industry. They vary in size from small units used for gas recovery (such as those on a crude distillation overhead system) to fairly large complex machines used for recycle gas streams and for transporting natural gas. Engineers are frequently required therefore to assess the horsepower of these machines and their capability to handle various streams. This item describes a method used to determine horsepower and proceeds with the following steps:

Step 1. Obtain the capacity and the properties of the gas to be handled. Fix the ultimate (discharge) pressure level.
Step 2. From the machine data sheet ascertain the number of stages.
Step 3. Estimate the brake horsepower from the expression:

$$\begin{aligned} \text{BHP} &= 22 \times (\text{compression ratio/stage}) \times \text{number of stages} \\ &\quad \times \text{capacity (in cuft/day} \times 10^6) \times \text{F} \end{aligned}$$

where

F = 1.0 for 1 stage
1.08 for 2 stages
1.10 for 3 stages

Ratio/Stage = $\sqrt{\ }$ ratio for two stages and $\sqrt[3]{\ }$ ratio for three stages.
Step 4. Check the estimate with Figure 18.28. The use of these graphs is self explanatory.
Step 5. Confirm actual suction conditions and compression ratio required (discharge pressure).
Step 6. Calculate compression ratio/stage.
Step 7. Calculate 1st stage discharge pressure. This will be suction pressure times compression ratio per stage from Step 6.
Step 8. Allow about 3% for inter-stage pressure drop then calculate second stage discharge pressure. Check that overall compression ratio/stage is close to that calculated for Step 6.

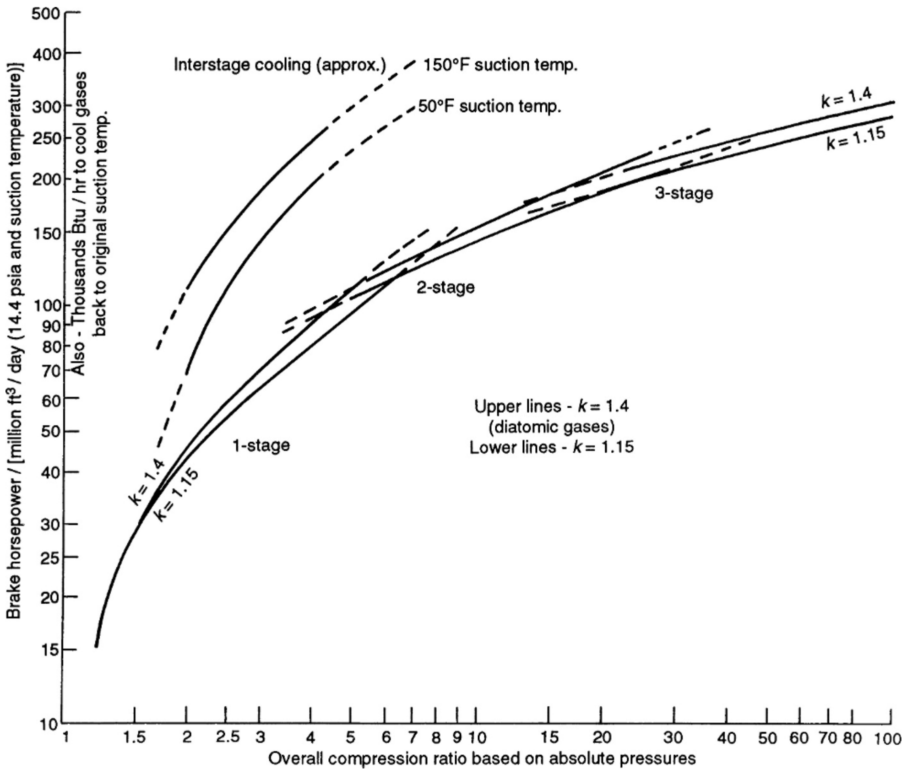


Figure 18.28. An estimate of brake horsepower per mm cfd for reciprocating compressors.

Step 9. Calculate the 'K' value of the gas. 'K' value is C_p/C_v of the gas. If the gas is a mixture of components, 'K' value may be calculated as the sum of each component multiplied be each of their 'K' values given in Table 18.A.1 in the Appendix. Alternatively for a good approximation data in Figure 18.29 may be used.

Step 10. Calculate discharge temperature from 1st stage using Figure 18.30. Assume some inter-cooling (or calculate inter-cooling from plant data) and fix 2nd stage discharge temperature using also using Figure 18.30.

Step 11. Calculate the compressibility factor Z at suction and discharge from the expression

$$\rho_v = \frac{\mathrm{MP}}{T \times Z \times 10.73}$$

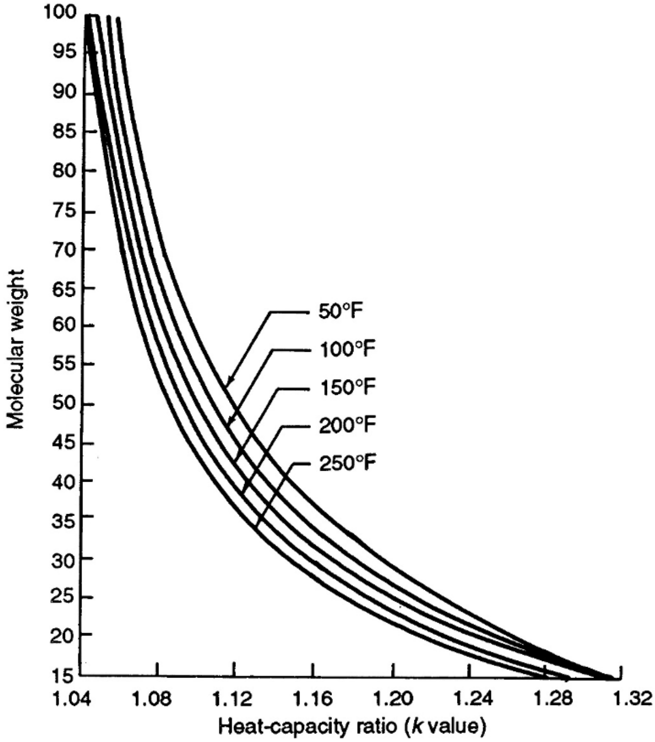


Figure 18.29. Approximation of 'K' from mole weights.

where

ρ_V = gas density in lbs/cuft at condition
T = °Rankine (°F + 460°F)
Z = compressibility factor
M = mole weight
P = pressure in psia

Use average value at suction and discharge for each stage.

Step 12. Read off BHP/mm cfd at the compression ratio/stage (from Step 7) and 'K' from Step 9 for each stage.

Step 13. Calculate BHP per stage from the expression:

$$\text{Bhp} = (\text{bhp/mmcfd}) \times \frac{\text{P}_\text{L}}{14.4} \times \frac{T_\text{S}}{T_\text{L}} \times Z_\text{ave} \times \text{mmScf/D}$$

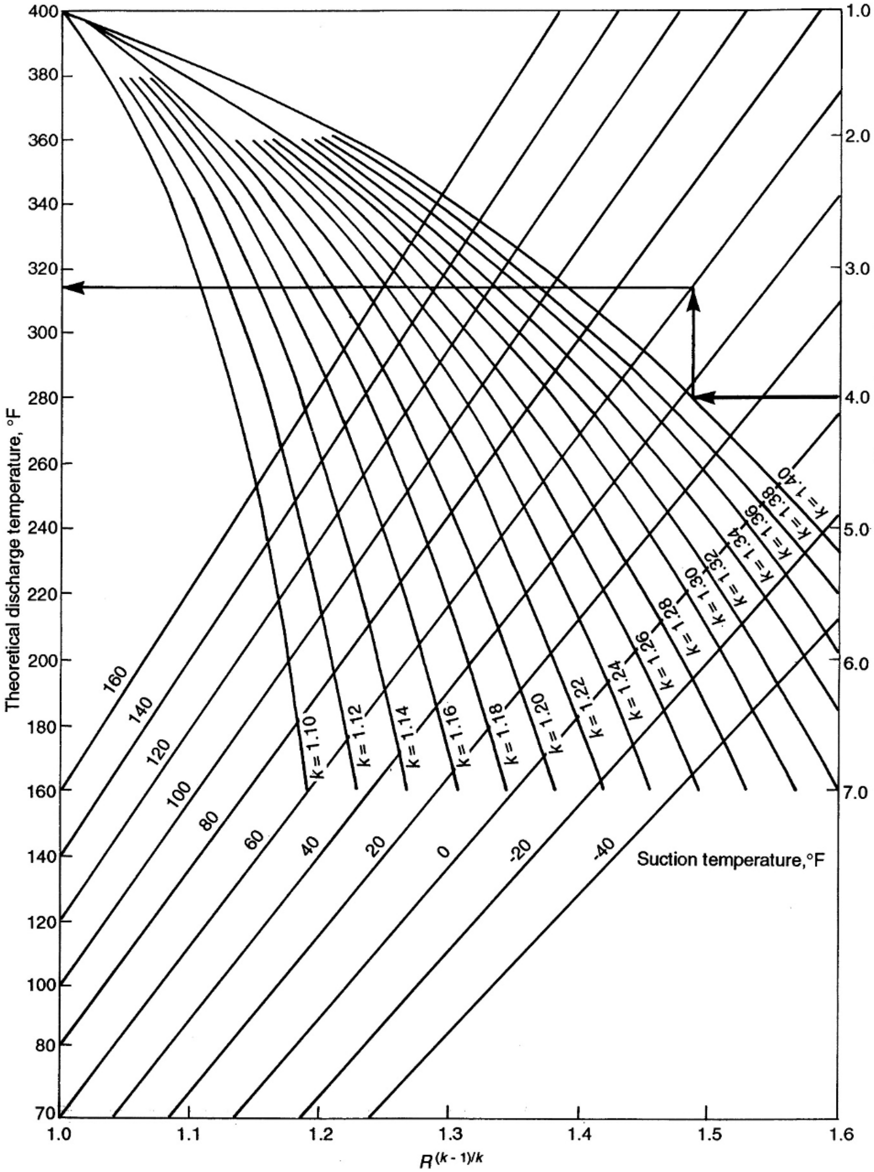


Figure 18.30. Determination of discharge temperature for reciprocating compressors.

where

Bhp/mmcfd = From Figure 18.29
P_L = pressure base used in contract psia
T_S = intake temperature °R
T_L = temperature base used in contract °R (usually 520°R)

Step 14. Brake horsepower for the machine is the sum of the BHP calculated for each stage in Step 13 above.

Reciprocating compressor controls and inter-cooling

A reciprocating compressor is a constant displacement type compressor. It compresses the same volume of gas to the same pressure level without regard to whether the gas is hydrogen or butane. This characteristic makes them desirable for use in services where the gas will have a widely varying composition. In some cases when an extremely low density gas will be compressed, a reciprocating compressor may be more economical than a centrifugal compressor, even though the flow rate may be very high, due to the large number of stages required for the centrifugal.

Reciprocating compressors are widely used in process services where the flow rates are too small for centrifugal compressors. These units can be obtained with integral or coupled electric motor in sizes from a few HP to 12,000 HP and separate or integral gas drivers varying in sizes from 100 HP to 5,500 HP.

A range of air-cooled light duty compressors is available for intermittent service. They range in size from 1/4 to about 100 HP at pressures up to 300 psig and are usually single acting. A primary process use of such equipment is for starting air compressors on gas engine driven machines. Reciprocating compressors can be designed to handle intermittent loads efficiently. This is done by using cylinder unloaders such as clearance pockets or suction valve lifters. Power losses are low at part load operation with these devices.

Reciprocating parts and pulsating flow present several engineering problems. The foundation and piping system must be constructed to withstand the vibrations produced by the compressor. The pulsating flow produced by the compressor must be dampened by the use of properly engineered suction and discharge bottles. These problems do not arise with the use of other types of compressors.

Reciprocating compressor control

Control of the compressor to prevent driver overload can be accomplished with clearance pockets, suction valve lifters, a throttling valves in the suction line, or a control

valve in a bypass around the compressor. A hand operated bypass without cooler is usually furnished inside the block valves for start-up purposes.

1. Clearance pockets

Of the above types of regulation, control by opening fixed clearance pockets gives the smoothest and most efficient control within its range of application. It has the following advantages:

- Minimizes the intake pulsation as the gas flow is not reversed in the intake lines to the cylinder.
- Results in lower bearing loads as all inertia loads are cushioned.
- Results in very efficient part load operation. When the gas compressed into the pockets is expanded, it follows the adiabatic line of compression and results in little power loss.

Clearance control has the following disadvantages which sometimes completely eliminates it from consideration:

- When low ratios of compression are combined with high suction pressure, clearance pockets of sufficient size to unload the compressor cannot physically be installed in the machine.
- Clearance control is designed for one set of pressure conditions and any variation in either suction or discharge pressure affects the amount of unloading accomplished by a given pocket.
- Condensable corrosive gases sometimes cause corrosion and liquid slugging problems.

2. Suction valve lifters

Suction valve lifters are the other type of internal unloading devices for compressor cylinders and have characteristics that make them applicable when clearance control is not. Suction valve lifters completely unload their end of the cylinder whenever they are opened, regardless of the pressure. They do result in increased bearing loads due to unbalanced inertia forces. Suction line pulsation may increase because the single acting cylinder may excite a different frequency in the gas.

3. Suction throttle valve

A throttle valve in the suction line should be considered only for small reciprocating compressors. For large size machines, the suction valve cannot give tight enough shut off to permit unloading the compressor for starting.

4. Bypass control

External bypass control around the compressor is applicable to all sizes of compressors. It results in a loss of power since the full compressor capacity must be compressed to and delivered at the full discharge pressure before being bled back to suction pressure. Care must be taken with this type of control to ensure that the bypassed gas is cooled sufficiently to prevent increasing the discharge temperature. This type of

control is preferred for installations up to several hundred horsepower because of its smoothness and lack of complexity. Individual machines can be shutdown for large process variations.

5. Variable speed reciprocating compressors
With a variable speed driver, cylinder control can usually be eliminated and speed control used to obtain desired process conditions. However, start-up unloading must be furnished, usually consisting of a hand operated bypass within the machine, or cylinder block valves. On turbine driven reciprocating compressors economics usually dictate that the compressor be run at constant speed and that cylinder controls or system bypasses be used to obtain the required control.

Reciprocating compressor inter-cooling
Inter-cooling for multistage compressors is advisable whenever there is a large adiabatic temperature rise within the cylinder and the cylinder discharge temperature would exceed 350°F. When inter-cooling is employed, the inlet temperature to the higher stage should be as close to the cooling water temperatures as practical. On standard commercial air inter-coolers, approach temperatures of 15–20°F are commonly used. Cooling to first stage inlet temperature is usually economical on process gas compressors.

Inter-cooling is employed for two basic reasons:

1. For mechanical reasons whereby discharge temperature must be limited to 350°F for lubrication purposes.
2. An economic reason as inter-cooling will save from 3% to 5% of the required BHP.

In general, on process compressors handling low "*n*" value gases inter-cooling is not employed unless the temperature limitation is exceeded. On high "*n*" value di-atomic gas mixtures, such as air, inter-cooling is the rule above about a 4 compression ratio and ambient temperature at suction.

In general, cooling water for electric driven compressors can be any water available, including salt water. (If the compressor is tied into a plant having gas engine driven compressors, the electric driven machine should be tied into the closed system). The cooling water should be available at a minimum pressure of 25 psig.

For estimating purposes, cooling water temperature rise across the cylinders can be taken as 15°F and that across the inter and after coolers can be taken as 15°F. For estimating purposes, cooling water requirements are as follows:

Jacket water cooling	500 Btu/BHP/hr
Inter cooler	1,000 Btu/BHP/hr
After cooler	1,000 Btu/BHP/hr

Use motor rating for number of horsepower required.

If the discharge temperature of the gas does not exceed 180°F, it is common practice to eliminate cooling water on the cylinder and operate with cooling passages which are filled with oil. Any jacketed cylinder must be filled with some fluid to ensure even temperature distribution.

Specifying a reciprocating compressor

As in the case of the centrifugal compressor all data necessary to give a precise requirement for the duty and performance required of a reciprocating compressor must be given in the specification sheet. Much of this data is the same as that given in a specification for a centrifugal compressor (discussed earlier in this chapter). For completeness however all the items in a reciprocating compressor are included below:

Title block

This requires the item to be identified by item number and its title. The number of units that the specification refers to is also given here. For a centrifugal compressor this will normally be just one as very seldom is a spare machine required. This may not be so in the case of a reciprocating compressor.

Normal and rated columns

More often than not the conditions and quantities required to be handled will vary during the operation of the machine. The two columns therefore will be completed showing the average normal data in the first column and the most severe conditions and duty required by the compressor in the second. The severe conditions in column two are for a continuous length of operation not instantaneous peaks (or troughs) that may be encountered.

Gas

The composition and gas stream identification must be included as part of the process specification. Usually the composition of the gas is listed on a separate sheet as shown in the example. Note in many catalytic processes that utilize a recycle gas the composition of the gas will change as the catalyst in the process ages. Thus it will be necessary to list the gas composition at the SOR and at the EOR.

The compressor may also be required to handle an entirely different gas stream at some time or other. This too must be noted. For example in many petroleum refining processes a recycle or make up compressor normally handling a light predominantly hydrogen gas is also used for handling air or nitrogen during catalyst regeneration, purging, and start-up.

Volume flow

This is the quantity of gas to be handled stated at 14.7 psia and 60 F.

Weight flow

This is the weight of gas to be handled in either lbs/min or lbs/hr.

Inlet conditions

In the case of multistage compressors the conditions for each stage must be shown. Where inter stage cooling is used the effect must be reflected in the conditions specified.

Pressure: This is the pressure of the gas at the inlet of the compressor stage in psia. Note if inter cooling is used this pressure must include the inter cooler pressure drop.

Temperature: This is the temperature of the gas at the inlet of the compressor stage—after the inter cooler if applicable.

Mole weight: The mole weight of the gas is calculated from the gas composition given as part of the specification.

C_p/C_v: This is the ratio of specific heats of the gas again obtained from the gas mole wt and Table 18.A.1 in the Appendix.

Compressibility factor (z): use the value at inlet conditions calculated as shown in step 7 of item on horsepower calculation for reciprocating compressors.

Inlet volume: This is the actual volume of gas at the conditions of temperature and pressure existing at the compressor stage inlet. Thus:

$$\text{ACFM} = \frac{\text{SCFM} \times 14.7 \times (\text{inlet temp F} + 460)}{(60\text{ F} + 460) \times \text{inlet press psia}}$$

Discharge conditions

Pressure: This is the pressure at each stage outlet and is quoted in either psia or psig.
Temperature: This is estimated for each stage using Figure 18.30.
C_p/C_v*:* This will be the same as in the inlet.

Approximate driver horsepower

The brake horsepower for the reciprocating compressor is calculated using the method described earlier. This is *Brake Horsepower* and includes an allowance for mechanical inefficiencies. The approximate minimum driver horsepower is 1.1 $\times$ Brake

Horsepower, but the approximate driver HP will be calculated using the inefficiencies for leakage, seals, etc as for centrifugal compressors.

The reminder of the spec sheet contains all the essential data and requirements that may affect the duty and performance of the compressor. Much of this is self explanatory; however there are some items that require comment. These are:

1. Most compressor installations today are under an open sided shelter with a small overhead gantry crane assembly for maintenance.
2. Usually the lube and seal oil assemblies have their own pump and control systems. Consequently even if the compressor itself is to be steam driven there may still be need to give details of utilities for the ancillary equipment.
3. Details of the gas composition is essential for any development of the compressor. This is listed on the last page of the specification together with any notes of importance concerning the machine and its operation.

An example calculation for a specification sheet follows:

Example calculation

A hydrotreater make up compressor is required to handle a gas stream such as to provide the unit with 260 SCF per barrel of feed of pure hydrogen. The composition of the gas varies as follows:

Mole %	Start of Run	End of Run
H_2	74.9	65.80
C_1	14.17	19.31
C_2	5.85	7.97
C_3	2.43	3.31
iC_4	1.13	1.54
nC_4	1.00	1.36
C_5S	0.52	0.71
Total	100.00	100.00

The fresh feed throughput is fixed at 30,000 BPSD (barrels per stream day). It is proposed to use 3 × 60% machines of which one will be standby and turbine driven. The inlet pressure of the gas is 50 psig at a temperature of 80 F. The gas is to be delivered at a pressure of 600 psig and 100 F. Prepare a process specification for reciprocating compressors to meet these requirements.

1.0 Calculating volume flows.

$$\text{SOR conditions Total flow required} = \frac{260}{0.749}$$

$$= 347 \text{ Scf of GAS per Bbl of feed.}$$

$$= \frac{347 \times 30{,}000}{24 \times 60} = 7{,}229 \text{ Scf/Min}$$

$$\text{EOR conditions Total flow required} = \frac{260}{0.658}$$
$$= 395 \text{ Scf/Bbl}$$
$$= \frac{395 \times 30{,}000}{24 \times 60} = 8,229 \text{ Scf/Min}$$

Volume flow per machine:

$$\text{SOR} = 7,229 \times 0.6 = 4{,}337$$
$$\text{EOR} = 8{,}229 \times 0.6 = 4{,}937$$

2.0 Calculate mole wt of gas.

	MW	SOR mole%	SOR wt Factor	EOR mole%	EOR wt Factor
H_2	2	74.9	149.8	65.8	131.6
C_1	16	14.17	226.7	19.31	309.0
C_2	30	5.85	175.5	7.97	239.1
C_3	44	2.43	106.9	3.31	145.6
iC_4	58	1.13	65.5	1.54	89.3
nC_4	58	1.00	58.0	1.36	78.9
C_5s	72	0.52	37.4	0.71	51.1
Total		100.0	644.3	100.00	1,044.6

SOR gas mole wt = 6.44   EOR gas mole wt = 10.44

3.0 Weight of gas lbs/min per machine

One mole of any gas occupies 378 cf at 60 F and 14.7 psia. Then

$$\text{For SOR Conditions moles/min of gas per machine} = \frac{4{,}337}{378} = 11.5$$

and

$$\text{lbs/min} = 11.5 \times 6.44$$
$$= 74.06 \text{ lbs/min}$$

$$\text{For EOR Conditions moles/min of gas per machine} = \frac{4{,}937}{378} = 13.06$$

and

$$\text{lbs/min} = 13.06 \times 10.44$$
$$= 136.30 \text{ lbs/min}$$

4.0 Inlet conditions

Inlet pressure = 50 psig = 65 psia.

Required outlet pressure = 600 psig = 615 psia.

$$\text{Overall compression ratio} = \frac{615}{65} = 9.46$$

This will be a two stage compressor. *Note:* At this level of compression in reciprocating compressors the compression ratio should not exceed 4:1 for any stage.

$$\text{Compression ratio per stage} = \sqrt{9.46} = 3.07.$$
$$\text{Discharge pressure stage 1} = 65 \times 3.07 = 199.6 \text{ psia}$$

Allowing 2 psi for the pressure drop across the inter cooler the suction pressure of stage 2 is 197.6 call it 198 psia.

Check the compression ratio of stage 2:

Required discharge pressure = 615 psia

$$\text{compression ratio} = \frac{615}{198} = 3.1$$

which is close to the originally predicted of 3.07.

5.0 Calculate ratio C_p/C_v.

		SOR		EOR	
	C_p/C_v	mole%	Factor	mole%	Factor
H_2	1.4	74.9	1.049	65.8	0.921
C_1	1.3	14.17	0.184	19.31	0.251
C_2	1.22	5.85	0.071	7.97	0.097
C_3	1.14	2.43	0.028	3.31	0.038
iC_4	1.11	1.13	0.013	1.54	0.017
nC_4	1.11	1.00	0.011	1.36	0.015
C_5's	1.09	0.52	0.006	0.71	0.008
Total		100.0	1.362	100.00	1.347

C_p/C_v SOR gas = 1.362
C_p/C_v EOR gas = 1.347

6.0 Calculate inlet ACFM per stage

SOR.

Inlet volume for 1st stage:

$$\begin{aligned}\text{ACFM} &= \frac{\text{Scf/min} \times 14.7 \times \text{ inlet temp R}}{(60 + 460) \times \text{ Inlet press psia}}\\ &= \frac{4{,}337 \times 14.7 \times 540}{520 \times 65}\\ &= 1{,}019 \text{ cf/min}\end{aligned}$$

Inlet volume for 2nd stage: (Inter cooled to 100 F)

$$\begin{aligned}\text{ACFM} &= \frac{4{,}337 \times 14.7 \times 560}{520 \times 198} \\ &= 347 \text{ cf/min}\end{aligned}$$

EOR.

Inlet volume for 1st stage:

$$\begin{aligned}\text{ACFM} &= \frac{4{,}937 \times 14.7 \times 540}{520 \times 65} \\ &= 1{,}159 \text{ cf/min}\end{aligned}$$

Inlet volume for 2nd stage:

$$\begin{aligned}\text{ACFM} &= \frac{4{,}937 \times 14.7 \times 560}{520 \times 198} \\ &= 488.5 \text{ cf/min}\end{aligned}$$

7.0 Calculate inlet compressibility factor (Z).

$$\text{Z} = \frac{\text{MW} \times P_i}{T_i \times \rho \times 10.73}$$

where

$$\rho = \frac{\text{wt/min}}{\text{ACFM}}$$

SOR conditions

$$\begin{aligned}\text{1st stage } Z &= \frac{6.44 \times 65}{540 \times 0.0727 \times 10.73} \\ &= 0.994 \\ \text{2nd stage } Z &= \frac{6.44 \times 198}{560 \times 0.213 \times 10.73} \\ &= 0.991\end{aligned}$$

EOR conditions

$$\begin{aligned}\text{1st stage } Z &= \frac{10.44 \times 65}{540 \times 0.1176 \times 10.73} \\ &= 0.996 \\ \text{2nd stage } Z &= \frac{10.44 \times 198}{560 \times 0.345 \times 10.73} \\ &= 0.997\end{aligned}$$

8.0 Determine discharge temperature

From Figure 18.30.

For SOR Conditions: 1st stage Comp Ratio = 3.07
$C_p/C_v = 1.362$
Suction Temp = 80 F
Discharge Temp read as 270 F
2nd stage Comp Ratio = 3.1
$C_p/C_v = 1.362$
Suction Temp = 100 F
Discharge Temp read as 299 F

For EOR Conditions: 1st stage Comp Ratio = 3.07
$C_p/C_v = 1.347$
Suction Temp = 80 F
Discharge Temp read as 262 F
2nd stage Comp Ratio = 3.1
$C_p/C_v = 1.347$
Suction Temp = 100 F
Discharge Temp read as 292 F

9.0 Compressibility factors for discharge conditions

SOR Conditions:

1st stage ACFM on discharge (before inter cooler).

$$= \frac{4{,}337 \times 14.7 \times 730}{520 \times 200 \,\text{(neglect IC pressure drop)}}$$

$$= 447.5 \text{ cf/min}$$

$$\rho = \frac{74.1}{447.5} = 0.166$$

$$Z = \frac{6.44 \times 200}{730 \times 0.166 \times 10.73}$$

$$= 0.991$$

2nd stage ACFM on discharge.

$$= \frac{4{,}337 \times 14.7 \times 759}{520 \times 615}$$

$$= 151.3 \text{ cf/min}$$

$$\rho = \frac{74.1}{151.3} = 0.490$$

$$Z = \frac{6.44 \times 615}{759 \times 0.49 \times 10.73}$$
$$= 0.992$$

EOR conditions.

These are calculated in the same way as those above and give the following results:

1st stage $Z = 0.995$
2nd stage $Z = 0.995$

10.0 Approximate driver horsepower.

Use the expression:

BHP = 22 × (Comp Ratio/stage) × No of stages × Capacity × Factor F
Comp ratio/stage = 3.07
No of stages = 2
Capacity per machine in mm cf/day at suction temperature
Factor for 2 stage machine = 1.08

For SOR conditions:

$$\text{BHP} = 22 \times 3.07 \times 2 \times (0.6 \times 10.81) \times 1.08$$
$$= 946$$

For EOR conditions:

$$\text{BHP} = 22 \times 3.07 \times 2 \times 7.38 \times 1.08$$
$$= 1{,}077$$

Use the efficiency factors as given in Figure 18.27 for centrifugal compressors. In this case there will be a gear assembly between compressor and driver. Use the efficiency of this as 97%. Thus:

	SOR	EOR
BHP	946	1,077
Gear losses (3% of BHP)	29	33
leakage	10	10
seal	35	35
Bearings	35	35
Driver HP	1,055	1,190

ITEM No. C101 A&B TITLE Hydrotreater Recycle Gas Compressor

Number of units required 3 (2 + 1 spare)

	Normal				Rated			
GAS (see attached composition)	75% H_2				65.8% H_2			
VOL. flow scf/min	4337				4937			
WEIGHT lb/min	74.1				136.3			
INLET CONDITIONS (each stage)	stage 1	stage 2	stage 3	stage 4	stage 1	stage 2	stage 3	stage 4
Pressure psia	65	198*			65	198*		
Temperature °F	80	100*			80	100*		
Mol weight	6.44				10.44			
C_p/C_v	1.362				1.347			
Compressibility factor	0.994	0.996			0.996	0.997		
Inlet vol acf/min	1019	347			1159	394		
DISCHARGE CONDITIONS (each stage)								
Pressure psia	198	615			198	615		
Temperature °F	270	299			262	292		
C_p/C_v	1.362				1.347			
Compressibility factor	0.991	0.992			0.995	0.995		
APPROX. DRVIER HORSEPOWER	1055				1190			

COMPRESSOR SERVICE REQUIRED: *After intercooling

Length of uninterrupted service: 8000 hours

Type of compressor: Lubricated Yes

Non Lubricated

Discharge RV setting (each stage) 250 and 750 psig

Figure 18.31. An example of a process specification for a reciprocating compressor.

Compressor drivers, utilities, and ancillary equipment

This item covers details on various compressor drivers, the utilities associated with operating the compressors and their ancillary equipment.

ITEM No. C 101 A&B TITLE Hydrotreater Recycle Gas Compressor

General Data and Requirements

Enclosure:

Open to weather ______

Under shelter Yes

In building ______

Corrosiveness and remarks concerning gas ______

None

Type of driver: Motor: 2 normal operating

Steam turbine:

Condensing ______

Non-condensing Spare machine

Utilities:

Power:

Voltage ______

Cycle ______

Phase ______

Steam:

Inlet: Pressure (psig) 600

Temperature (°F) 710

Condensing exhaust

Pressure (psia) N/A

Temperature (°F) N/A

Non-condensing exhaust

Pressure (psig) 50

Temperature (°F) (by vendor)

Cooling water: Pressure 60 psig: Temperature 40 °F

Allowable temperature rise 30 °F

Materials of construction:

Cylinder CS Type By vendor

Piston Ni.Cr Type By vendor

Type of shaft seal: Labyrinth

Figure 18.31. (Cont.)

Compressor drivers

Table 18.24 gives a listing of the more common types of compressor drivers. It provides some of the data that would influence the choice of the driver. The most common drivers by far in a process plant are the electric motor and the steam turbine.

ITEM No. C 101 A&B TITLE Hydrotreater Recycle Gas Compressor

General Data and Requirements (cont.)

Gas Composition

Mol %	Start of run	End of run
H_2	74.9	65.80
C_1	14.17	19.31
C_2	5.85	7.97
C_3	2.43	3.31
iC_4	1.13	1.54
nC_4	1.00	1.36
c_5s	0.52	0.71
Total	100.00	100.00

Remarks and notes:

1. Vendor to provide:

Intercoolers: Yes

Aftercooler: Yes

Dampeners: Yes

Flushing and sealing oil systems: Yes

2. If motor driven Re acceleration required class A

3. Suction line: Size 6" RTG 300# RF

Discharge line: Size 4" RTG 600# RF

Figure 18.31. (Cont.)

For very large machines as encountered in handling natural gas the gas turbine or gas engine become the more prominent prime mover.

Sizing drivers

As a basic rule drivers are sized for the most severe duty required of the compressor plus a factor as an operating contingency. In general the most severe duty is that design case which has the highest suction temperature, the maximum ratio of specific heats, the lowest suction pressure, the highest required discharge pressure, and the gas molecular weight which gives the highest HP. The driver rated horsepower shall

Table 18.24. Comparison of compressor drivers

Driver	HP range	Available speed, RPM	Efficiency %	Common applications
Synchronous motor	100–20,000	3,600	90–97	Reciprocating compressors
Induction motor	1–15,000	3,600	86–94	All types of compressors
Wound rotor induction motor	–	–	–	Normally not used
Steam engine	10–4,000	400–140	60–80	All types of rotary equip.
Steam turbine	10 to 2,000	2,000 to 15,000	50 to 76	Centrif, axial and recip.
Combustion gas turbine	3,000 to 35,000	10,000 to 3,600	19 to 24 see note 1 below	All types of compressors (except recip)
Gas and oil engines	100 to 5,000	1,000 to 300	35 to 45	Reciprocating compressors

Note 1: The efficiency given here does not include for waste heat recovery. With WHR the efficiency can be increased to between 28 and 35%.

therefore be greater than:

$$\text{Driver Brake HP} = \frac{\text{Max compressor BHP @ the most severe duty}}{\text{Mechanical efficiency of the power transmission}}.$$

The mechanical efficiency in this case includes for energy losses for bearings, seals, lube oil etc, in the case of centrifugal compressors and gears in the case of reciprocating compressors.

Electric motor drivers

Squirrel cage motors are preferred for this type of duty. These may be drip proof open type where the location is not a fire or explosion hazard. Where it is required that the units must be explosion or fire proof these motors must be totally enclosed type. In sizing the motor, efficiencies for squirrel cage motors up to 450 HP given in item 4.6 for pumps may be used. The following Table 18.25 is used for motors above 500 HP.

The driver rated brake horsepower is the compressor horsepower times a load factor divided by a service factor. Normally the load factor is 10% and a service factor for an enclosed squirrel cage motor is 1.0 and 1.15 for an open type.

Example calculation
Calculate the Operating Load and the Connected Load for the driver of a 4,000 HP centrifugal compressor (includes Leakage, Seal, and Bearing losses). A gear is used and this has a 97% efficiency. The load factor is 10% and the motor is open type squirrel cage with a service factor of 1.15. There will be a normal operating unit and a spare, both motor driven.

Table 18.25. Motor efficiencies

	Motor efficiencies full load @ percent of		
Motor Rated HP	50	75	100
500	91.4	93.1	93.4
1,000	92.1	93.8	94.1
	92.4	94.1	94.4
2,000	92.7	94.4	94.7
2,500	92.9	94.6	94.9
3,000	93.0	94.7	95.0
3,500	93.0	94.7	95.0
4,000	93.1	94.8	95.1
4,500	93.1	94.8	95.1
5,000	93.2	94.9	95.2

$$\text{Minimum required driver HP} = \frac{4{,}000 \times 1.1}{0.97}$$
$$= 4{,}536$$
$$\text{Driver nameplate rating} = \frac{4{,}536}{1.15} = 3{,}944$$

call it 4,000 HP.

Connected load for the motor is:

$$\begin{aligned}\text{Motor nameplate rating} \times 1.15 &= 4{,}000 \times 1.15\\ &= 4{,}600 \text{ HP (rated HP)}\\ &= \frac{4{,}600 \times .746}{0.951 \text{ (@ 100\% load)}}\\ &= 3{,}608 \text{ kW}\end{aligned}$$

$$\begin{aligned}\text{There are two units then Total connected load} &= 3{,}608 \times 2\\ &= 7{,}216 \text{ kW}\end{aligned}$$

Operating load for the motor is:

$$\frac{4{,}000}{0.97} = 4{,}124 \text{ HP}$$
$$\%\text{ load} = \frac{4{,}124}{4{,}600} = 90\% \text{ (use 75\% eff)}$$
$$\text{Operating load} = \frac{4{,}124 \times 0.746}{0.948}$$
$$= 3{,}245 \text{ kW.}$$

Table 18.26. Steam Turbine Efficiencies

	Adiabatic efficiencies %	
	Inlet pressures (psig.)	
Driver BHP	900	100
500	48	59
800	53	64
1,000	56	67
1,200	58	68
1,500	60	71
2,000	63	73
2,500	65	74
3,000 and up	67	76

Steam turbine drivers. Next to the motor drivers steam turbines are the most common form of drivers for rotary equipment in general and compressors in particular. The two most common types of theses are turbines that exhaust to a lower pressure but the exhaust steam is not condensed and those in which the exhaust steam is condensed. Normally the latter is only used in the case of large driver horsepower 5,000 and above. It is far more expensive than the non condensing type as the exhaust is normally sub atmospheric in pressure and of course the cost of the condenser must be included.

Steam turbine approximate efficiencies are listed in Table 18.26.

These efficiencies are based on the exhaust pressure of the steam being 50 psig for non condensing type and 2″ Hg Abs for the condensing type.

Determining the rated horsepower of the steam turbine driver follows closely to the method for motor horsepower. First determine the minimum horsepower required of the turbine. Thus:

Step 1. Determine the *minimum* driver horsepower by multiplying the compressor BHP by 1.1.

Step 2. Now the turbine will deliver the normal HP at the normal speed. A contingency in the form of additional speed is added to the driver capability. This will be controlled in practice by a steam governor. This contingency is usually 5% above normal speed.

Step 3. Horsepower capability varies as the cube of the speed. Thus the Rated horsepower of the turbine will be:

$$\text{Rated HP} = \text{Minimum HP} \times (1.05)^3$$

Step 4. The amount of steam that will be used calculated by the change in enthalpy of the inlet steam to the outlet steam at constant entropy. The change in enthalpy for the two conditions is read from the steam Mollier diagram.

Step 5. The theoretical steam rate is:

$$\frac{2{,}544}{\text{Inlet enthalpy} - \text{Outlet enthalpy (in Btu/lb)}}$$

This figure divided by the turbine efficiency gives the steam rate in lbs/BHP/hr.

Example calculation

Calculate the turbine horsepower requirements and the theoretical steam rates to drive a 4,000 BHP centrifugal compressor. No gears are included in this case. Steam is available at 650 psig and 760 F. The steam will exhaust into the plant's 125 psig header.

$$\begin{aligned}\text{Minimum driver horsepower} &= 4{,}000 \times 1.1\\ &= 4{,}400 \text{ BHP}\\ \text{Rated turbine HP @ 105\% speed} &= 4{,}400 \times (1.05)^3\\ &= 5{,}094 \text{ HP.}\end{aligned}$$

$$\begin{aligned}\text{Enthalpy of steam @ 650 psig and 760 F} &= 1{,}390 \text{ (entropy 1.62)}\\ \text{Enthalpy of steam @ 125 psig} &= 1{,}225 \text{ (entropy 1.62)}\end{aligned}$$

Difference in Enthalpy = 165

Efficiency of turbine (from Table 18.26) = 67%.

$$\begin{aligned}\text{Theoretical Steam Rate} &= \frac{2{,}544}{165 \times 0.67}\\ &= 23 \text{ lbs/BHP/hr.}\end{aligned}$$

Gas turbine drivers. These items of equipment are the most expensive and because they require a high capital investment their use can only be justified as compressor drivers where the continual load on the compressor is also very high. These drivers therefore are met mostly in the natural gas industry. They are used extensively in recompressing natural gas after treating for dew point control or desulfurizing.

The thermal efficiencies of gas turbines are low (about 16–20%) but it is common practice to use the exhaust gases which are usually at a temperature of above 800 F in waste heat recovery. This involves exchanging the waste heat of the exhaust gases

Table 18.27. Gas turbine sizes and data

HP rating @ 80, and 1,000 ft	Fuel consumption LHV lbs/Hp-hr	Exhaust Flow #/sec	Exhaust Temp F	RPM
430	1.25	10.3	950	19,250
1,000	0.66	11.1	960	19,500
1,080	0.63	13.7	860	22,300
1,615	0.84	23.6	1,000	13,000
2,500	0.76	43.0	795	9,000
3,800	0.75	53.0	900	8,500
5,500	0.76	77.3	945	5,800
7,000	0.8	101.0	938	5,500
8,000	0.65	102.0	935	5,800
9,000	0.70	130.0	850	5,000
10,000	0.59	123.5	805	6,000
12,000	0.65	160.0	720	4,750
13,500	0.62	187.0	800	4,860
15,000	0.61	188.0	835	4,860
24,000	0.61	258.0	850	3,600

with boiler feed water to generate steam or to preheat a process stream, for example distillation. Table 18.27 gives some turbine sizes and data. It should be noted that considerable development work is continuing in the field of gas turbines and consequently the data given here may be subject to revision or updating.

To obtain the gas turbine rated horsepower for a specific compressor duty follows closely the same calculation route as the steam turbine. Thus:

Step 1. Obtain the *minimum driver horsepower* by multiplying the compressor Brake horsepower by 1.05 (BHP includes Seals, Leakage, etc).

Step 2. The rated turbine horsepower is the minimum driver HP divided by the gear efficiency.

Step 3. The horsepower of the turbine selected must equal or slightly exceed the horsepower calculated in Step 2. This HP must be corrected for site conditions as shown in Step 4.

Step 4. The HP's given in Table 18.27 are at an ambient temperature of 80 F and at an elevation of 1,000 ft. correction for any specific site is given by the following expression:

$$\text{SITE HP} = \text{Quoted HP}\,(1.00 + \text{A} \times 10)^{-2}(1.00 - \text{B} \times 10)^{-2} \times (1.00 - \text{C} \times 10)^{-2} \times \frac{\text{Site Atmos Press}}{14.7}$$

where

A = Temp adjustment of % per F
B = Inlet press loss % per ins water gauge.
C = Discharge press loss % per ins water gauge.

'A' will be plus for ambient temperatures above 80°F and minus for ambient temperatures below 80°F.

Ancillary equipment

Reciprocating compressor dampening facilities. Dampening facilities are used in conjunction with reciprocating compressors to smooth out the pulsation effect of the compressor action. These facilities are simply in line bottles sized larger than the gas line which cushion the gas motion. These are essential to minimize expensive piping designs that would be necessary without them. Calculating the size of these bottles is important in the design of the compressor facilities.

The following calculation technique is used to determine the size of new dampers or to evaluate the adequacy of an existing facility. This calculation is described by the following steps:

Step 1. From compressor data sheet obtain cylinder diameter and stroke dimensions.
Step 2. Calculate the swept volume per cylinder using the expression:

$$\frac{\pi D^2}{4} \times S$$

where

D = cylinder diameter
S = stroke

Step 3. Knowing the suction and discharge pressures the pulsation bottle capacity (both suction and discharge) is obtained from Figure 18.32 in terms of a multiple of swept volume.

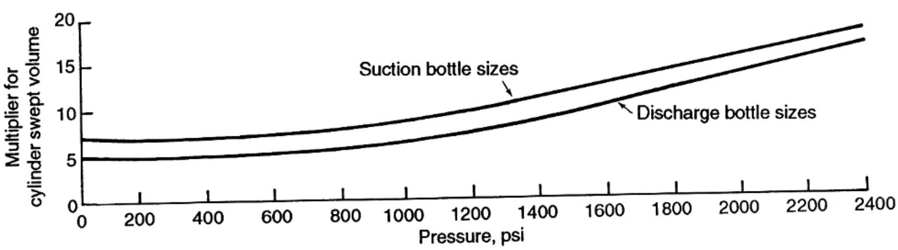


Figure 18.32. Dampener bottle sizing.

Step 4. Use the rule of thumb that pulsation bottle diameter equals 1½ times the compressor cylinder diameter. Calculate the suction and discharge bottle length.

Example calculation
To determine the dimensions of the compressor pulsation bottle of a reciprocating compressor having a 6″ diameter cylinder and a stroke of 15″. The compressor delivers 3.0 mm Scf/D gas at a suction pressure of 100 psia and 100°F, and a discharge pressure of 1,200 psia.

The cylinder diameter is 6″ and stroke is 15″

$$\begin{aligned}
\text{then swept volume} &= \pi/4 \times 6^2 \times 15 \\
&= 424 \text{ cu.ins} \\
\text{capacity of machine} &= 3 \text{ mmScf/D} \\
\text{in a mm CF/D} &= \frac{3 \times 14.7 \times 560}{520 \times 100} \\
&= 0.475 \text{ mm ACFD} \\
&= 330 \text{ ACF/min} \\
&= 570240 \text{ AC ins/min} \\
\text{machine speed} &= \frac{570240}{424} = 1{,}345 \text{ RPM}
\end{aligned}$$

From Figure 18.32:

Suction bottle size should be 7 × swept volume (at 100 psia) discharge bottle size should be 7 × swept volume (at 1,200 psia) = 2,968 cu ins or 1.718 cu.ft

As a rule of thumb diameter of bottle should be 1½ × cylinder diameter = 0.75 ft (9″) length = 2,968/63.6 = 47 ins or 4 ft.

18.4 Heat exchangers

Type and selection of heat exchangers

Heat exchange is the science that deals with the rate of heat transfer between hot and cold bodies. There are three methods of heat transfer, they are:

- Conduction
- Convection
- Radiation

In a heat exchanger heat is transferred by conduction and convection with conduction usually being the limiting factor. The equipment used in heat exchanger service is designed specifically for the duty required of it. That is, heat exchange equipment

cannot be purchased as a stock item for a service but has to be designed for that service.

The types of heat exchange equipment used in the process industry and their selection for use are as follows:

The shell and tube exchanger

This is the type of exchanger most commonly used in a process plant. It consists of a bundle of tubes encased in a shell. It is inexpensive and is easy to clean and maintain. There are several types of shell and tube exchangers and some of these have removable bundles for easier cleaning. The shell and tube exchanger has a wide variety of services that it is normally used for. These include vapor condensation (condensers), process liquid cooling (coolers), exchange of heat between two process streams (heat exchangers), and reboilers (boiling in fractionator service). Most of this chapter will be dedicated to the uses and design specification of the shell and tube exchanger.

The double pipe exchanger

A double pipe exchanger consists of a pipe within a pipe. One of the fluid streams flows through the inner pipe while the other flows through the annular space between the pipes. The exchanger can be dismantled very easily and therefore be easily cleaned. The double pipe exchanger is used for very small process units or where the fluids are extremely fouling. Either true con-current or counter current flows can be obtained but because the cost per square foot is relatively high it can only be justified for special applications. The following table gives the heat transfer area for various pipe lengths and diameters:

No of tubes	Shell size, ins	Tube size, ins	Surface area, sqft		
			10 ft	20 ft	30 ft
1	2	1	5.8	11.0	16.3
1	3	1.5	10.9	20.9	30.9
1	4	2	13.7	26.1	38.5

Extended surface or fin tubes

This type of exchanger is similar to the double pipe but the inner pipe is grooved or has longitudinal fins on its outside surface. Its most common use is in the service where one of the fluids has a high resistance to heat transfer and the other fluid has a

low resistance to heat transfer. It can rarely be justified if the equivalent surface area of a shell and tube exchanger is greater than 200–300 sqft.

Finned air coolers

These are the more common type of air coolers used in the process industry. Air cooling for process streams gained prominence during the early 1950s. In a great many applications and geographic areas they had considerable economic advantage over the conventional water cooling. Indeed today it is uncommon to see process plants of any reasonable size without air coolers.

Air coolers consist of a fan and one or more heat transfer sections mounted on a frame. In most cases these sections consist of finned tubes through which the hot fluid passes. The fan located either above or below the tube section induces or forces air around the tubes of the section.

The selection of air coolers over shell and tube is one of cost. Usually air coolers find favor in condensing fractionator overheads to temperatures of about 90–100°F and process liquid product streams to storage temperatures. Air coolers are widely used in most areas of the world where ambient air temperatures are most times below 90°F. At atmospheric temperatures above 100°F humidifiers are incorporated into the cooler design and operation. The cost under these circumstances is greatly increased and their use is often not justified.

In very cold climates the air temperature around the tubes is controlled to avoid the skin temperature of the fluid being cooled falling below a freezing criteria or in the case of petroleum products its pour point. This control is achieved by louvers installed to recirculate the air flow or by varying the quantity of air flow by changing the fan pitch.

Box coolers

These are the simplest form of heat exchange. However they are generally less efficient, more costly and require a large area of the plant plot. They consist of a single coil or “worm” submerged in a bath of cold water. The fluid flows through the coil to be cooled by the water surrounding it. The box cooler found use in the older petroleum refineries for cooling heavy residuum to storage temperatures. Modern day practice is to use a tempered water system where the heavy oil is cooled on the shell side of a shell & tube exchanger against water at a controlled temperature flowing in the tube side. The water is recycled through an air cooler to control its temperature to a level which will not cause the skin temperature of the oil in the shell & tube exchanger to fall below its pour point.

Direct contact condensers

In this exchanger the process vapor to be condensed comes into direct contact with the cooling medium (usually water). This contact is made in a packed section of a small tower. The most common use for this type of condenser is in vacuum producing equipment. Here the vapor and motive steam for each ejector stage is condensed in a packed direct contact condenser. This type has a low pressure drop which is essential for the vacuum producing process.

General design considerations

Basic heat transfer equations

The following equations define the basic heat transfer relationships.

These equations are used to determine the overall surface area required for the transfer of heat from a hot source to a cold source.

The overall heat transfer equation
The usual heat transfer mechanism are conduction, natural convection, forced convection, condensation, and vaporization. When heat is transferred by these means the overall equation is as follows:

$$Q = UA(\Delta t_m)$$

where

Q = Heat transferred in Btu/hr
U = Overall heat transfer coefficient, Btu/hr/sqft/°F
A = Heat transfer surface area, sqft.
Δt_m = Corrected log mean temperature difference, °F

The overall heat transfer coefficient U
This coefficient is the summation of all the resistances to the flow of heat in the transfer mechanism. These resistances are the resistance to heat transfer contained in the fluids, the resistance caused by fouling, and the resistance to heat transfer of the tube wall. The resistance to the flow of heat from the liquid outside the tube wall is measured by the film coefficient of that fluid. The resistance of the flow of heat from the fluid inside the tube is similarly the film coefficient of the inside fluid. These film coefficients are products of dimensionless numbers which include:

- The Reynolds Number
- The Graetz Number
- The Grashof Number

- The Nusselt Number
- The Peclet Number
- The Prandtl Number
- The Stanton Number

The format of these numbers and their use are found in all standard text books on heat transfer. For example: Kern's *Process Heat Transfer*, and McAdams *Heat Transmission*.

These resistances are defined therefore by the following expression:

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{1}{h_i} \times \frac{A_o}{A_i} + \frac{1}{h_w} + (rf)_o + (rf)_i \times \frac{A_o}{A_i}$$

where

U_o = overall heat transfer coefficient based on outside tube surface, in Btu/hr/sqft/°F.
h = The film coefficient in Btu/hr/sqft/°F.
rf = fouling factors in $\dfrac{1}{\text{Btu/hr/sqft/}^\circ\text{F}}$
h_w = Heat transfer rate through tube wall in Btu/hr/sqft/°F.
A = Surface area in sqft

subscripts "o" and "i" refer to outside surface and inside surface, respectively.

Flow arrangements

The two more common flow paths are Con-current and Counter-current. In Con-current flow both the hot fluid and the cold fluid flow in the same direction. This is the least desirable of the flow arrangement and is only used in those chemical processes where there is a danger of the cooling fluid congealing, subliming, or crystallizing at near ambient temperatures.

Counter-current flow is the most desirable arrangement. Here the hot fluid enters at one end of the exchanger and the cold fluid enters at the opposite end. The streams flow in opposite directions to one another. This arrangement allows the two streams exit temperatures to approach one another.

Logarithmic mean temperature difference Δt_m

In either counter-current or con-current flow arrangement the log mean temperature difference used in the overall heat transfer equation is determined by the following

expression:

$$\Delta t_{\mathrm{m}} = \frac{\Delta t_1 - \Delta t_2}{\ln \dfrac{\Delta t_1}{\Delta t_2}}$$

The Δt's are the temperature differences at each end of the exchanger and Δt_1 is the larger of the two. In true counter-current flow the Δt_{m} calculated can be used directly in the overall heat transfer equation. However such a situation is not common and true counter-current flow rarely exists. Therefore a correction factor needs to be applied to arrive at the correct Δt_{m}. These are given in Figure A9.1 in the Appendix.

Fluid velocities and pressure drops

Film coefficients are a function of fluid velocity, density (vapor), and viscosity (liquids). Within limits increasing the velocity of a fluid reduces its resistance to heat transfer (i.e., it increases its heat transfer coefficient). Increasing the fluid velocity however increases its pressure drop. An economic balance needs to be sought therefore between the cost of heat transfer surface and pumping cost. This exercise should be undertaken to find a pay-out balance of 2–4 years. This exercise has been done many times and the following data is considered a reasonable balance between velocity and pressure drop for some common cases:

	Tube side		Shell side	
	Velocity, ft/sec	Press drop, psi	Velocity, ft/sec	Press drop, psi
Non-viscous liquids	6–8	10	1.5–2.5	10
Viscous liquids	6	20	3.0 max	15–20
Clean cooling water	6–8	10–15	–	–
Dirty cooling water	3 min	10+	–	–
Suspended solids in	2–3 min	10	1.5 min	15 liquids. (Note 1)
Gases and Vapors	$\dfrac{100}{\sqrt{\text{Gas density}}}$ max	3–5	–	3
Condensing vapors	–	–	–	3–5

Note 1: Normally erosion by suspended solids in liquids occurs at velocities of above 6 ft/sec.

For condensing steam pressure drop is usually not critical but a minimum steam pressure drop is desirable. Allowable steam velocities in tubes are as follows:

Pressure	Velocity ft/sec
Below atmospheric	225
Atmos to 100 psig	175
Above 100 psig	150

Choice of tube side versus shell side

There are no hard and fast rules governing which fluid flows on which side in a heat exchanger. Much is left to the discretion of the individual engineer and his or her experience. There are some guidelines and these are as follows:

(i) *Tube side flow*

Fouling liquids.
Tube cleaning is much easier than cleaning the outside of the tubes. Also fouling can be reduced by higher tube side velocities.

Corrosive fluids. It is cheaper to replace tubes than shells and shell baffles so as a general rule corrosive fluids are put on the tube side. There are exceptions and a major one are those corrosive fluids that become more corrosive at high velocities. An example of this are naphthenic acids, which are present in some crude oils and their products.

High pressure. Fluids at high pressure are usually put on the tube side as only the tubes, tube sheet, and channel need to be rated for high pressure in the unit design. This cheapens the overall cost of the exchanger.

Suspended solids. Fluids containing suspended solids should whenever possible be put to flow tube side. Shell side flows invariably have “dead spaces” where solids come out of suspension and build up to cause fouling.

Cold boxes. These are exchangers used in cryogenic processes where condensing of a vapor on one side of the exchanger is accompanied by boiling of a liquid on the other side. The condensing fluid is preferred on the tube side. Better control of the refrigerant flow is accomplished by the level control across the shell side.

(ii) *Shell side flow*

Available pressure drop. Shell side flows generally require lower pressure drop than tube side. Therefore if a system is pressure drop limiting it should be routed to the shell side.

Condensers. Condensing vapors should flow on the shell side wherever possible. The larger free area provided by the shell side space permits minimum pressure drop and higher condensate loading through better film heat transfer coefficients.

Large flow rates. In cases where both streams are of a similar nature with similar properties the stream with the largest flow rate should be sent to the shell side where the difference in flow rates are significant. The shell side provides more flexibility in design by baffle arrangements to give the best heat transfer design criteria.

Boiling service. The boiling liquid as in the case of reboilers, waste heat recovery units and the like should be on the shell side of the exchanger. This allows space for the proper disengaging of the vapor phase and provides a means of controlling the system by level control of the liquid phase.

Types of shell and tube exchangers

Figure 18.33 gives some of the more common arrangements in shell and tube exchanger design. The arrangements shown here are all one shell pass and one or two tube passes. Equipment with more than two tube passes (up to five) is also fairly common particularly in petroleum refining. Shell arrangements are however left at one if at all possible. Where multi-pass shell side is required companies prefer to use complete exchangers in series or in parallel or both rather than making two or more shell passes using horizontal baffling in one exchanger.

Estimating shell and tube surface area and pressure drop

There are many excellent computer programs available that calculate exchanger surface area and pressure drops from simple input. The actual calculation when done manually is tedious and long. However to understand a little of the importance of the input required by these computer programs it does well to at least view a typical manual calculation. The one given here is for a shell and tube cooler with no change of phase for either tube side or shell side fluids.

The calculation follows these steps:

Step 1. Establish the following data by heat balances or from observed plant readings:

- The inlet and outlet temperatures on the shell side and on the tube side.
- The flow of tube side fluid and that for the shell side. It may be necessary to calculate one or the other from a heat balance over the exchanger.
- Calculate the duty of the exchanger in heat units per unit time (usually hours).
- Establish the stream properties for tube side and shell side fluids. The properties required are: SG, Viscosity, Specific heats, Thermal conductivity.

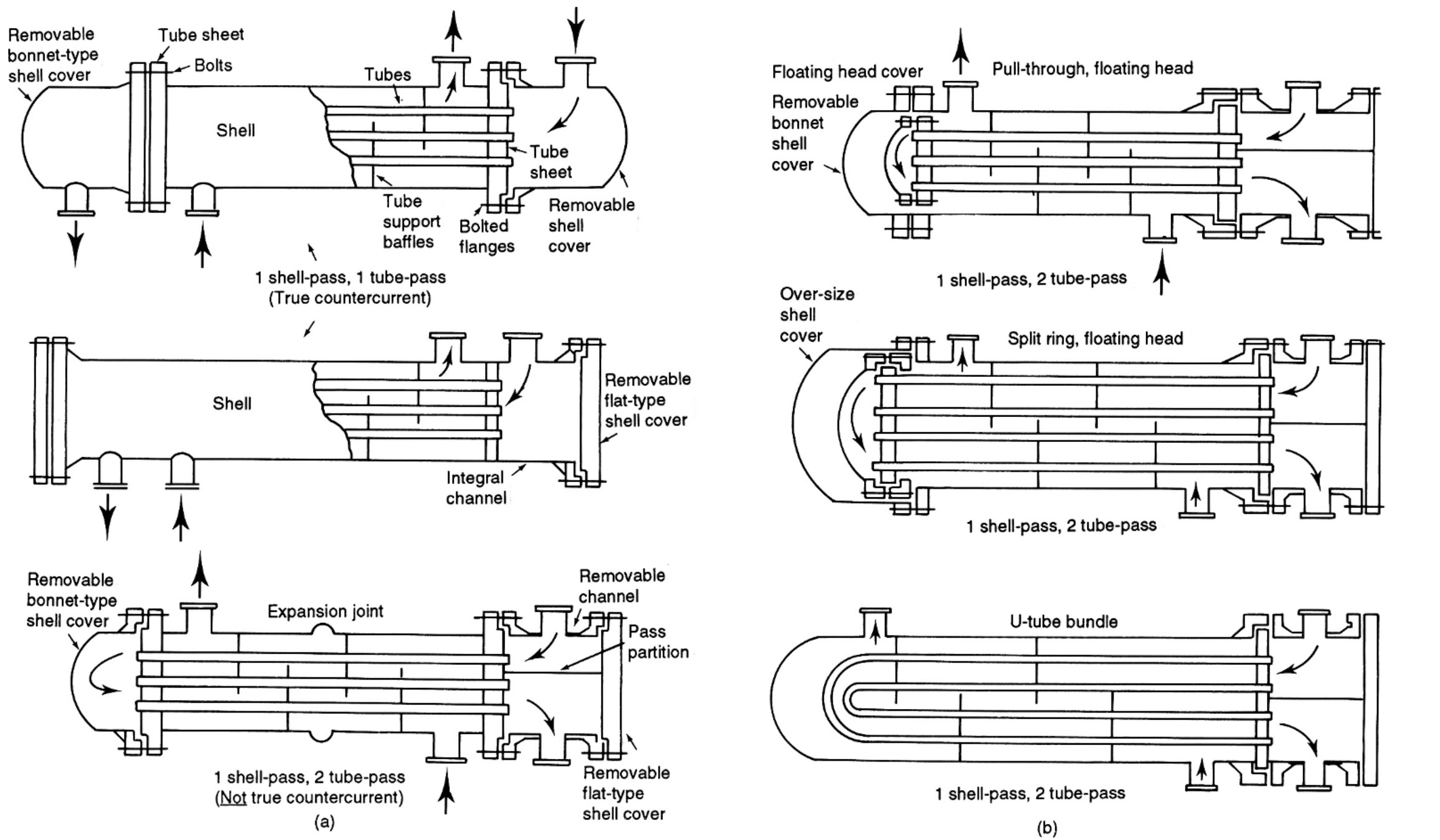


Figure 18.33. Some Common types of shell and tube exchangers. (a) Fixed tube sheet. (b) Removable bundle.

Step 2. Calculate the log mean temperature difference (Δt_m).

Assume a flow pattern (i.e. either co current or counter-current). Most flows will be a form of counter-current. Then show the temperature flow as follows:

Shell In	$\longrightarrow$	Shell Out
Tube out	$\longleftarrow$	Tube In
Temp difference Δt_1		Δt_2

The log mean temperature difference is then calculated using the expression:

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\log_e \dfrac{\Delta t_1}{\Delta t_2}}$$

This temperature needs to be corrected for the flow pattern, and this is done using the correction factors given in Figure 18.A.1. The use of these are self explanatory and are given in the figure.

Step 3. Calculate the approximate surface area.

From Table 18.A.2 in the Appendix select a suitable overall heat transfer coefficient U in Btu's/hr. sqft. F. Use the expression to calculate for 'A':

$$Q = UA\Delta t_m$$

where

Q = Heat transferred in Btu/hr. (the exchanger duty)
U = Overall heat transfer coefficient.
A = Exchanger surface area in sqft.
Δt_m = Log mean Temperature difference (corrected for flow pattern) in °F.

From the surface area calculated select the tube size and pitch. Usually ¾ ins on a triangular pitch for clean service and 1″ on a square pitch for dirty or fouling service. A single standard shell will hold about 4,100 sqft of surface per pass. Now most companies do not use multi-pass shells and prefer sets of shells in series if this becomes necessary. The 'norm' therefore are single pass shells each containing up to 4,100 sqft of surface.

Step 4. Calculate the tube side flow and the number of passes.

If it cannot be read from plant data calculate tube side flow in cuft/hr by heat balance. Select the tube gauge and length. The tube data are given in Table 18.A.3 in the Appendix and standard lengths of tubes are 16 and 20 ft. Calculate the number of selected tubes per pass from the expression:

$$Np = \frac{Ft \times 144}{3{,}600 \times At \times Vt}$$

where

Np = number of tubes per pass
Ft = tube side flow in cuft/hr
At = cross-sectional area of 1 tube
Vt = linear velocity in tube in ft/sec

See earlier item on 'Fluid Velocities and Pressure Drop' for recommended fluid velocities. The number of tube passes is arrived at by dividing the total surface area required by the total (external) surface area of the number of tubes per pass calculated above.

Step 5. Calculate tube side film coefficient corrected to outside diameter (h_{io}).
The tube side film coefficient may be calculated for water by the expression:

$$h_{io} = \frac{300 \times (V_t \times \text{tube i/d ins})^{0.8}}{\text{tube o/d ins}}$$

where

h_{io} = Inside film coefficient based on outside tube diameter in Btu/hr sqft °F.
V_t = linear velocity of water tube side in ft/sec.

For fluids other than water flowing tube side use the expression:

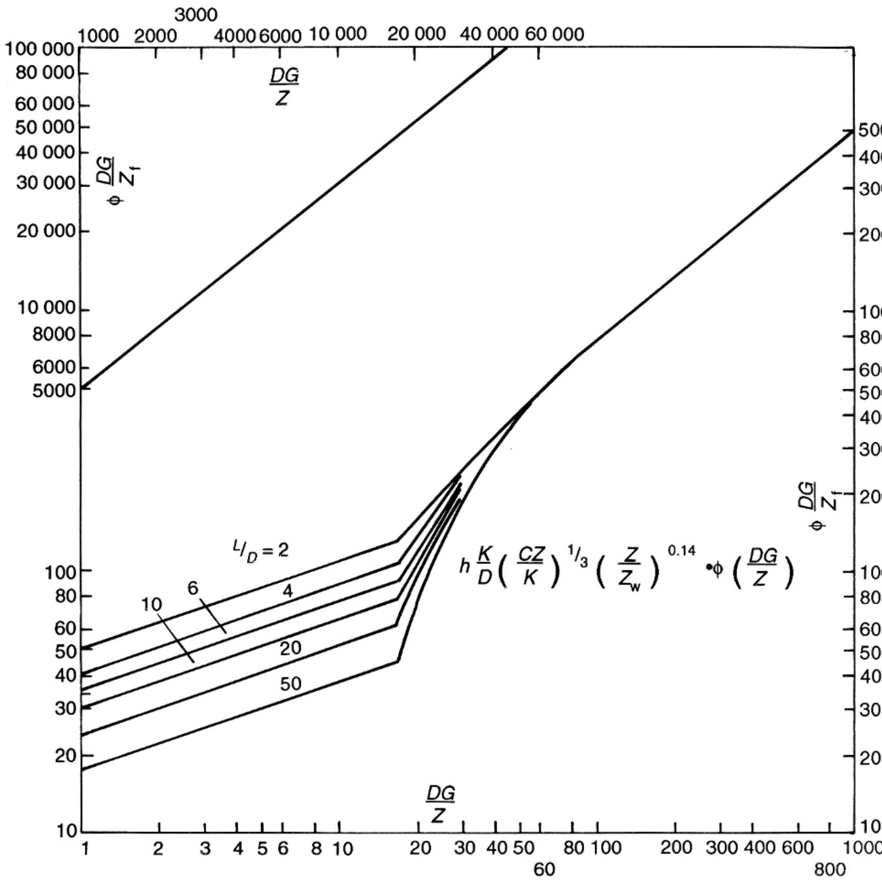
$$h_{io} = \frac{K}{D_o}(C\mu/K)^{1/6}(\mu/\mu_w) \cdot \phi(DG^{.14}/\mu)$$

where

h_{io} = inside film coefficient based on outside diameter in Btu/hr sqft °F.
K = thermal conductivity of the fluid in Btu/hr sqft (°F per ft). See Maxwell *Data Book on Hydrocarbons* or Perry *Chemical Engineers Handbook*
D = Inside tube diameter in ins.
D_o = Outside tube diameter in ins.
C = Specific heat in Btu/lb/°F.
G = mass velocity in lbs/sec sqft.
μ = Absolute viscosity Cps at average fluid temp.
μ_w = Absolute viscosity Cps at average tube wall temp.
$\phi(DG/\mu)$ = from Figure 18.34.

Step 6. Calculate shell side dimensions.
First determine the shell side average film temperature as follows:

$$\text{Inlet ave} = \frac{T_1 + T_2}{2} \quad \text{Outlet ave} = \frac{T_3 + T_4}{2}$$



h = Film coefficient (BTU/h/ft^2/ °F)
K = Thermal conductivity (BTU/ h ft^2/ (°F per ft)
L = Heated tube length (ft)
D = Inside tube diameter (in)
G = Mass velocity (lb/ s / ft^2)
C = Specific heat of fluid at av. fluid temp. (BTU/ lb/ °F)
Z = Absolute viscosity (cp) at av. temp. of fluid
Z_W = Absolute viscosity (cp) at av. tube wall temp.

Figure 18.34. Heat transfer inside tubes.

where

T_1 = Shell fluid inlet temperature.
T_2 = Tube outlet temperature.
T_3 = Shell outlet temperature.
T_4 = Tube inlet temperature.

Average shell side film temperature:

$$\frac{\text{Inlet ave} + \text{outlet ave}}{2}$$

Use this temperature to determine density and viscosity used in the shell side film coefficient calculations.

The shell diameter. Next calculate the diameter of the tube bundle and the shell diameter. For this use one of the following equations to calculate the number of tubes across the center line of the bundle:

1. For square pitch tube arrangement:
   $T_{cl} = 1.19\ (\text{number of tubes})^{0.5}$
2. For triangular pitch tube arrangement:
   $T_{cl} = 1.10\ (\text{number of tubes})^{0.5}$

Note these are *total* number of tubes; namely, those calculated in Step 4 times number of tube passes.

Set number of baffles and their pitch. The type of baffles usually used are shown in Figure 18.35. Disc and donut type baffles are only used where pressure drop available is very small and there is a pressure drop problem. Baffles on the bias are used in sq pitch tube arrangement and baffles perpendicular to the tubes are usual for triangular tube arrangements.

The minimum baffle pitch should not be less than 16% of the shell diameter. Pitch in this case is the space between two adjacent baffles. Normally 20% of shell i/d is used for the baffle pitch. The number of baffles is calculated from the expression:

$$N_B = \frac{10 \times \text{tube length}}{\text{baffle pitch}\,\% \times \text{diameter of shell.}}$$

Free area of flow between baffles. The space available for flow on the shell side is calculated as:

$$W = D_i - (d_o \times T_{cl})$$

where

W = space available for flow in sq ins.
D_i = shell inside diameter in ins.
d_o = tube outside diameter in ins.
T_{cl} = number of tubes across centerline.

The free area of flow between baffles is now calculated as follows:

$$A_f = W \times (B_p - 0.187)$$

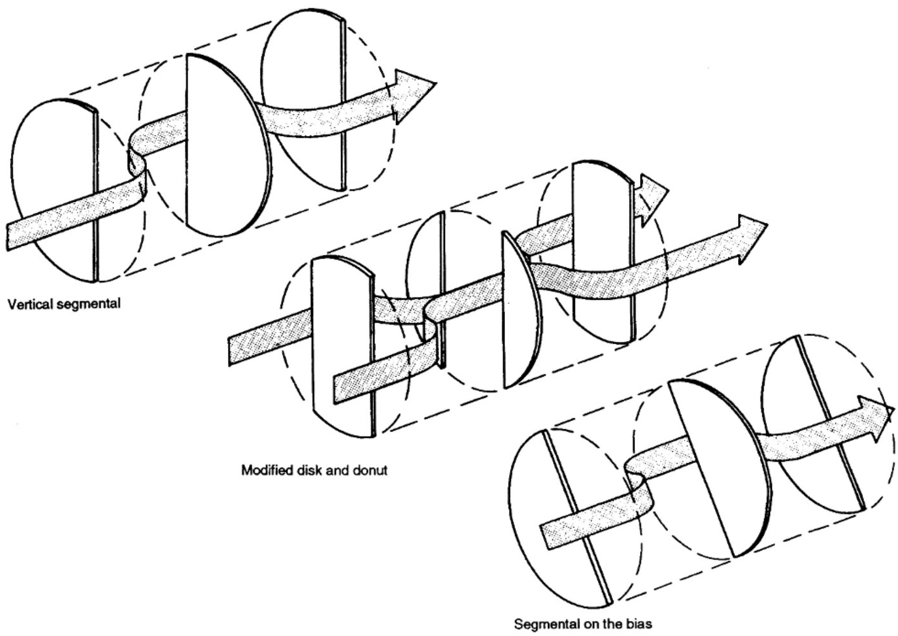


Figure 18.35. Types of baffles.

where

A_f = free flow area between baffles in sq ins.
B_p = baffle pitch in ins.

Step 7. Calculate the shell side film coefficient h_o.

The following expression is used to determine the outside film coefficient:

$$h_o = \frac{K}{d_o}(C\mu/K)^{1/3} \cdot \phi(d_o G_m/\mu_f) \cdot \frac{4P_b}{D}$$

where

h_o = outside film coefficient in Btu/hr sqft °F
G_m = maximum mass velocity in lbs/sec. sqft
d_o = outside tube diameter in ins
K = thermal conductivity.
C = specific heat of fluid in Btu/lb/F
μ_f = viscosity at mean film temperature in Cps
P_b = baffle pitch in ins
D = shell internal diameter in ins

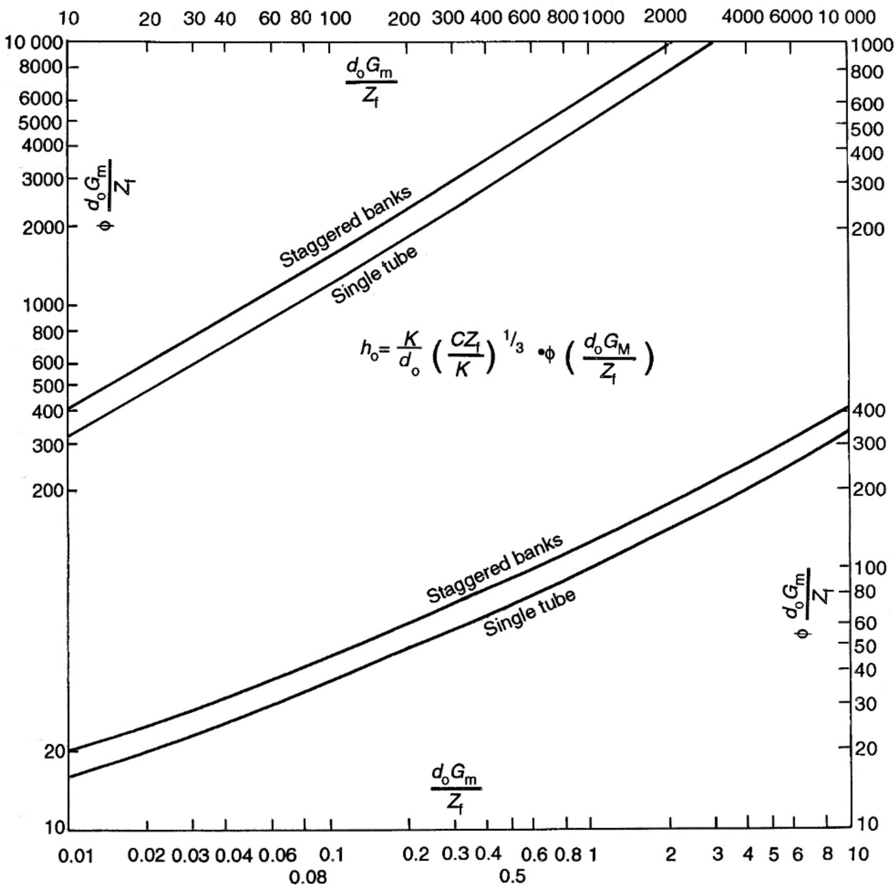


Figure 18.36. Heat transfer to fluids outside tubes.

$\phi(d_o G_m/\mu_f)$ is a function of the Reynolds number read from Figure 18.36. The Reynolds Number is:

$$\mathrm{Re} = \frac{d_o G_m}{\mu_f}$$

where G_m = lbs/sec sqft.

This film coefficient is corrected for the type of baffle and tube arrangement by multiplying it by one of the following factors:

For square pitch vertical to tube rows	0.50
square pitch on the bias	0.55
triangular tube pitch	0.70

Table 18.28. Thermal conductivity of tube metals

	K, Btu/hr.sqft. F/ft
Admiralty brass	64
Aluminum brass	58
Aluminum	117
Brass	57
Carbon steel	26
Copper	223
Cupronickel	41
Lead	20
Monel	15
Nickel	36
Red Brass	92
Type 316 alloy steel	9
Type 304 alloy steel	9
Zinc	65

Step 8. Calculate the *overall heat transfer coefficient U_o.*

The film coefficients calculated in steps 5 and 7 are now used in the expression:

$$\frac{1}{U_o} = \frac{1}{h_{io}} + r_{io} + \frac{1}{h_o} + r_o + r_w$$

where

U_o = overall heat transfer coefficient in Btu/hr.sqft.°F.

r_{io} and r_o = tube side and shell side fouling factors respectively in hr. sqft.°F/Btu. For clean tubes this is 0.001 as a sum of both factors.
r_w = Tube wall resistance to heat transfer in hr. sqft. °F/Btu, which is expressed as:

$$r_w = \frac{t_w \cdot d_o}{12 \times K \times (d_o - 2t_w)}$$

where

t_w = Tube wall thickness ins.
d_o = Outside tube diameter ins.
K = Thermal conductivity Btu's/hr sqft °F/ft. See Table 18.28.

Compare the calculated value of the overall heat transfer coefficient with the assumed one in step 3.
If there is agreement within ±10% then the calculated one will be used for revising the calculation for surface area and the other dimensions. If there is no agreement repeat the calculation using a new value for the assumed U.

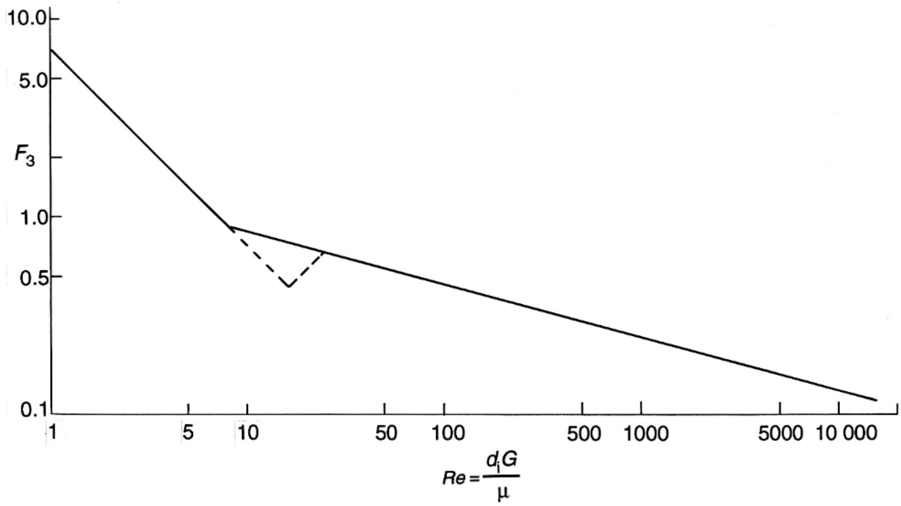


Figure 18.37. Pressure drop factor F_3 for flows inside tubes.

Step 9. Calculate tube side pressure drop.

Using the adjusted dimensional values from the calculated U_o, calculate the tube side pressure drop using one of the following equations:

$$\Delta P_t = .02 F_t \times N_p \times (V^2 + (0.158 L V^{1.73}/d_i)^{1.27})$$

For water only.

For fluids other than water use:

$$\Delta P_t = F_t \times N_p \times (\Delta P_{tf} + \Delta P_{tr})$$

where

$\Delta P_{tf} = F_3 \cdot \frac{L}{d_i} \cdot (\rho_m \times V^2/9,270) \cdot (\mu_w)^{.14}$
$\Delta P_{tr} = 3 \times (\rho_m \times V^2/9,270)$
F_3 = factor based on Reynolds number see Figure 18.37
ρ_m = density in lbs/cuft at mean fluid temperature.
μ_w = viscosity of fluid at tube wall temperature in Cps (use mean film temperature)
V = linear velocity in ft/sec.
F_t = pressure drop fouling factor as follows (dimensionless)

Tube OD	Tube metal	F_t
0.75	Steel	1.50
1.00	Steel	1.40
1.50	Steel	1.20
0.75	Ad Brass	1.20
1.00	Copper	1.15

The pressure drop figure calculated by these equations are for one unit. Where there are more than one shell in series multiply the figures by the number of shells.

Step 10. Calculate the shell side pressure drop.

Using the revised dimensions calculated in Step 8 the total shell side pressure drop is calculated using the following equation:

$$\Delta P_s = F_s(\Delta P_{sr} + \Delta P_{sf})$$

where

$\Delta P_{sf} = B_2 F_{sp} N_{tc} N_b (m \times V^2/9{,}270)$

$\Delta P_{sr} =$ pressure drop due to turns given by:

$$(N_b + 1) \cdot (3.5 - 2P_b/D) \cdot \frac{(m \times V^2)}{9{,}270}$$

$B_2 =$ Factor as follows:

Baffle position	*Tube layout*	B_2
Vertical	square	0.30
Bias @ 45°	square	0.40
Vertical	triangular	0.50

$F_{sp} =$ Factor based on Reynolds number. See Figure 18.38.
$N_{tc} =$ Number of tubes on center line.
$N_b =$ Number of shell baffles.
$P_b =$ Space between baffles ins.
$D =$ Shell i/d in ins.

The pressure drop calculated here is for one shell. If there are more than one shell in series then multiply these pressure drops by the number of shells.

Air coolers and condensers

Air cooling of process streams or condensing of process vapors is more widely used in the process industry than cooling or condensing by exchange with cooling water. The use of individual air coolers for process streams using modern design techniques has economized in plant area required. It has also made obsolete those large cooling towers and ponds associated with product cooling. This item of the chapter describes air coolers in general and outlines a method to estimate surface area, motor horsepower and plant area required by the unit.

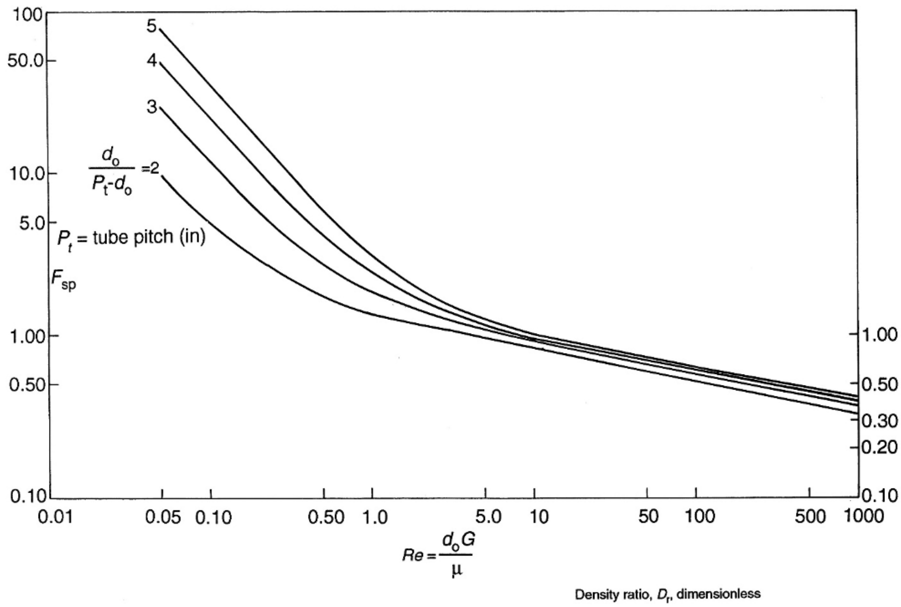


Figure 18.38. Pressure drop factor F_{SP} for flows across banks of tubes.

As in the case for shell and tube exchangers there are many excellent computer programs that can be used for the design of air coolers. The method given here for such calculation may be used in the absence of a computer program or for a good estimate of a unit. The method also emphasizes the importance of the data supplied to manufacturers for the correct specification of the units.

General description of air coolers/condensers

Figures 18.39 show the two types of air coolers used in the process industry. Both units consist of a bank of tubes through which the fluid to be cooled or condensed flows. Air is passed around the tubes either by a fan located below the tubes forcing air through the tube bank or a fan located above the tube bank drawing air through the tube bank. The first arrangement is called 'forced draft' and the second 'induced draft'.

Air in both cases is motivated by a fan or fans driven by an electric motor or a steam turbine or in some cases a gas turbine. The fan and prime driver are normally connected by a 'V' belt or by a shaft and gear box. Electric motor drives are by far the most common prime drivers for air coolers.

The units may be installed on a structure at grade or as is often the case on a structure above an elevated pipe rack. Most air coolers in condensing service are elevated above pipe racks to allow free flow of condensate into a receiving drum.

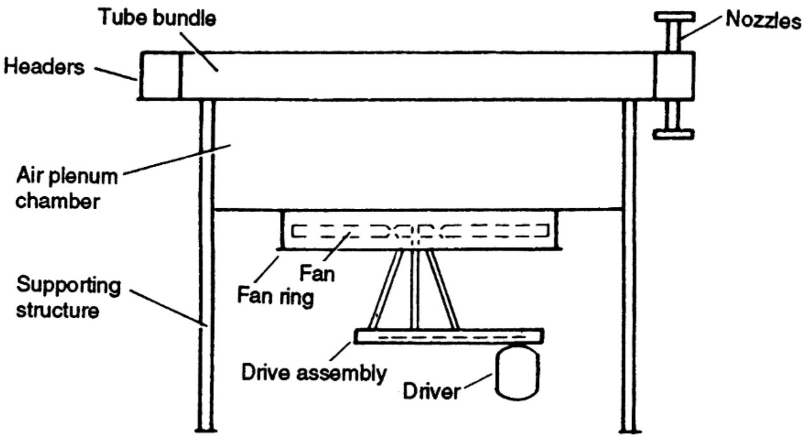


Fig A A Forced Air Flow Arrangement

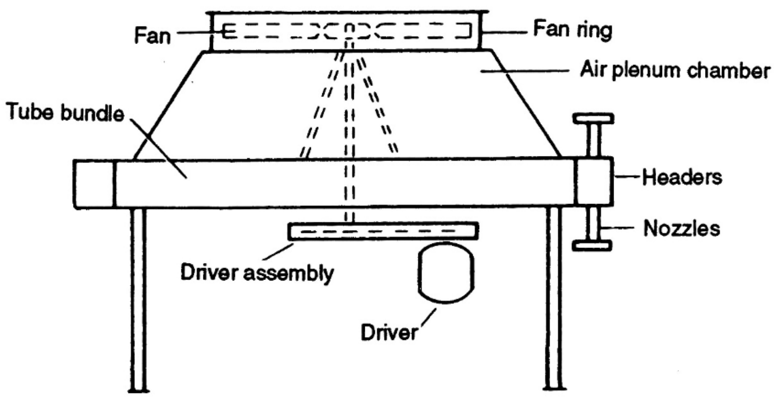


Fig B An Induced Air Flow Arrangement

Figure 18.39. Air coolers—(a) Forced draught. (b) Induced draught.

Thermal rating

Thermal rating of an air cooler is similar in some respects to that for a shell and tube described in the previous item. The basic energy equation

$$Q = U \Delta T A$$

is used to determine the surface area required. The calculation for U is different in that it requires the calculation for the air side film coefficient. This film coefficient is

usually based on an extended surface area which is formed by adding fins to the bare surface of the tubes. Thermal rating, surface area, fan dimensions and horsepower are calculated by the following steps:

Step 1. Calculate the heat duty and the tube side material characteristics.
Step 2. Calculate the log mean temperature for the exchanger.
Using the following equation determine the temperature rise for the air flowing over the tubes:

$$\Delta t_m = ((U_e + 1)/10)) \cdot ((\Delta t_m/2) - t_1))$$

where

Δt_m = air temperature rise °F
U_e = overall heat transfer coefficient assumed. (from Table 18.29).
Δt_m = mean tube side temperature °F
t_1 = inlet air temperature °F.

Calculate the log mean temperature difference (LMTD) as in Step 2 of previous item on shell and tubes.

Step 3. Determine an approximate surface area using the expression:

$$A_E = \frac{Q}{U_E \cdot \Delta t_m}$$

where

A_E = extended surface area in sqft.
Q = exchanger duty in Btu/hr.
U_E = overall heat transfer coefficient based on extended surface from Table 18.29.
Δt_m = log mean temperature difference corrected for number of passes in °F.

Step 4. Calculate the number of tubes from the expression:

$$N_t = \frac{A_E}{A_f \times L}$$

where

N_t = total number of tubes.
A_E = extended surface area in sqft.
A_f = extended area per ft of fin tube read from Table 18.30.
L = length of tube (30 ft is standard).

Step 5. Fix the number of passes (usually 3 or 4) and calculate the mass flow of tube side fluid using the expression:

$$G = \frac{\text{lbs/hr of tube side fluid} \times N_p \times 144}{N_t \times A_t \times 3{,}600}$$

Table 18.29. Some common overall transfer coefficients for air cooling

	1/2″ by 9 Fin ht by Fin/ins		5/8′ by 10 Fin ht by Fin/ins	
Service	U_e	U_o	U_e	U_o
Process water	95	6.5	110	5.2
Hydrocarbon Liquids				
Visc @ ave temp cps				
0.2	85	5.9	100	4.7
1.0	65	4.5	75	3.5
2.5	45	3.1	55	2.6
6.0	20	1.4	25	1.2
10.0	10	0.7	13	0.6
Hydrocarbon gasses				
@ Pressures psig				
50	30	2.1	35	1.6
100	35	2.4	40	1.9
300	45	3.1	55	2.6
500	55	3.8	65	3.0
1,000	75	5.2	90	4.2
Hydrocarbon condensers				
Cooling range 0°F	85	5.9	100	4.7
10°F	80	5.5	95	4.4
60°F	65	4.5	75	3.5
100+°F	60	4.1	70	4.2
Refrigerants				
Ammonia	110	7.6	130	6.1
Freon	65	4.5	75	3.5

U_e is transfer coefficient for finned surface.
U_o is transfer coefficient for bare tubes.

Table 18.30. Fin tube to bare tube relationships based on 1″ O/D tubes

Fin Ht by Fins/ins	1/2″ by 9		5/8″ by 10	
Area/ft Fin tube	3.8		5.58	
Ratio of Areas Fin/Bare Tube	14.5		21.4	
Tube Pitch ins	2Δ	2¼Δ	2¼Δ	2½Δ
Bundle Area sqft/ft (Note 1)				
3 Rows	68.4	60.6	89.1	80.4
4 Rows	91.2	80.8	118.8	107.2
5 Rows	114.0	101.0	148.5	134.0
6 Rows	136.8	121.2	178.2	160.8

Note 1: Bundle area is the external area of the bundle face area in sqft/ft.

where

G = Mass velocity in lbs/sec sqft.
N_p = Number of tube passes.
A_t = inside cross-sectional area of tube in sq ins

Step 6. Calculate the Reynolds Number for tube side using the expression:

$$\mathrm{Re} = \frac{d_i \cdot G}{\mu}$$

where

R_e = Reynolds Number (dimensionless)
d_i = Tube i/d in ins.
μ = Tube side fluid viscosity at average temperature in Cps.

Step 7. Calculate the inside film coefficient from the expression:

$$h_{io} = \frac{K}{D}(C\mu/K)^{1/3} \cdot (\mu/\mu_w)^{.14} \cdot \phi(DG/Z)$$

where

h_i = inside film coefficient in Btu/hr · sqft °F.
K = thermal conductivity of the fluid in Btu/hr · sqft (°F per ft). See Maxwell *Data Book on Hydrocarbons*.
D = inside tube diameter in ins.
C = specific heat in Btu/lb/°F.
G = mass velocity in lbs/sec sqft.
μ = absolute viscosity Cps at average fluid temp.
μ_w = absolute viscosity Cps at average tube wall temp.
$\phi(DG/\mu)$ = from Figure 18.34

Step 8. Calculate the mass velocity of air and the film coefficient on the air side thus:

$$\text{Weight of air} = \frac{Q}{C_{\text{Air}} \times \Delta t_{\text{Air}}}$$

where

Q = exchanger duty in Btu/hr
C_{Air} = specific heat of air (use 0.24)
Δt_{Air} = temperature rise of the air °F.

Face area of tubes A_f is calculated as follows:
Set the O/D of the tubes (usually 1″), length, fin size (usually 5/8″ @ 10 to the ins or 1/2″ @ 9 to the ins), Pitch (see Table 18.30), and number of tube rows (start with 3 or 4). Then face area is:

$$A_f = \frac{\text{Total extended surface area } A_E}{\text{External area per ft of bundle (from Table 18.30)}}$$

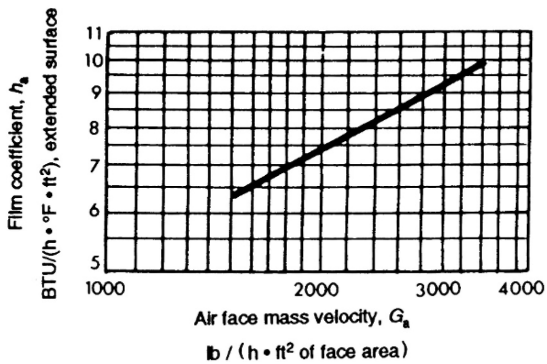


Figure 18.40. Air film coefficients.

Mass velocity of air is calculated from the expression:

$$G_a = \frac{\text{lbs per hour of air flow}}{\text{face area } A_f}$$

The film coefficient for the air side is read from Figure 18.40.

Step 9. Calculate the overall heat transfer coefficient as follows:

Area ratio of bare tube outside to finned outside is read from Table 18.30. Then factor to convert all heat flow resistance to outside tube diameter basis is:

$$F_t = \frac{A_r \times \text{tube o/d}}{A_t}$$

where

A_r = Area ratio
A_t = inside tube cross-sectional area sq ins.

Then:

$$\frac{1}{U_o} = \frac{1 + (r_t \times F_t) + r_w + 1}{h_i h_o}$$

where

r_t = inside fouling factor.
r_w = tube metal resistance (normally ignored.)

If the calculated U is within 10% of the assumed there will be no need to recalculate with a new assumed value for the U. The dimensions and data are adjusted however using the calculated value for U.

Step 10. Calculate the required fan area and the fan diameter as follows:

$$\text{Fan area} = \frac{0.4 \times \text{Face Area } A_f}{\text{Assumed Number of Fans}}$$

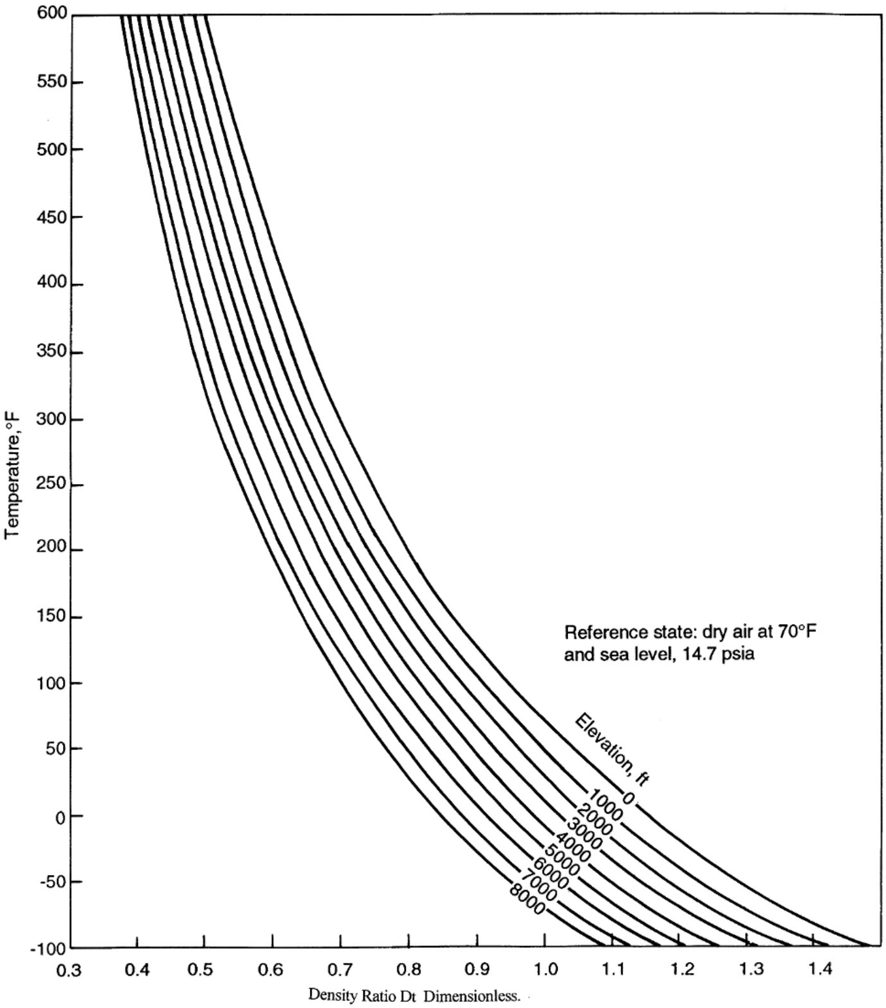


Figure 18.41. Relative density of air.

Begin by assuming 2 fans and continue with multiples of 2 until a reasonable fan diameter (about 10 to 12 ft) is obtained. On very large units fans can be maximized at 16 ft.

$$\text{Fan diameter} = \sqrt{}(\text{Fan area} \times 4/\pi)$$

Step 11. Calculate air side pressure drop and actual air flow in cuft/min.

$$\text{Average air temperature} = \frac{t_1 + t_2}{2}$$

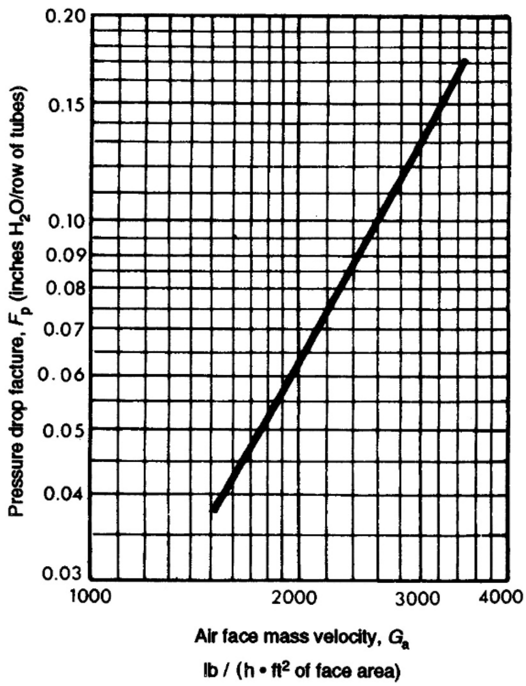


Figure 18.42. Pressure drop air side in inches of water.

From Figure 18.41

D_r = Relative density factor for air at elevations of site.

From Figure 18.43, ΔP_a = Pressure drop of air in ins of H_2O

$$\Delta P_a \text{ corrected} = \frac{\Delta P_a \times \text{No of rows}}{D_r}$$

Density of air at corrected ΔP_a

$$\frac{29}{(378 \times 14.7 \times T_2)/(T_1 \times (\text{Corr } \Delta P_a + 14.7))}$$

where $T_{1\&2}$ are absolute temperatures.
ACFM of air therefore is:

$$\frac{\text{lbs/hr of air}}{\text{Density} \times 60}$$

ΔP of air at the fan is obtained by the expression:

$$\Delta P_m = \left[\frac{\text{ACFM}}{(4{,}000(\pi d)/4)}\right]^2$$

in ins of water gauge.

Step 11. Calculate the fan horsepower as follows:

$$\text{Hydraulic HP} = \frac{\text{ACFM} \times \text{Density of air} \times \text{Diff head in Ft}}{33{,}000}$$

$$\text{Differential head} = \frac{\text{Total } \Delta P @ \text{ fan in ins } H_2O \times 5.193}{\text{Density}}$$

$$\text{Bhp} = \frac{\text{Hydraulic HP}}{\eta_r}$$

where η_r is the fan efficiency (usually 70%).

Condensers

In petroleum refining and most other chemical process plants vapors are condensed either on the shell side of a shell and tube exchanger, the tube side of an air cooler, or by direct contact with the coolant in a packed tower. By far the most common of these operations are the first two listed. In the case of the shell and tube condenser the condensation may be produced by cooling the vapor by heat exchange with a cold process stream or by water. Air cooling has overtaken the shell and tube condenser in the case of water as coolant in popularity as described in the previous item.

In the design or performance analysis of condensers the procedure for determining thermal rating and surface area is more complex than that for a single phase cooling and heating. In condensers there are three mechanisms to be considered for the rating procedure. These are:

- The resistance to heat transfer of the condensing film
- The resistance to heat transfer of the vapor cooling
- The resistance to heat transfer of the condensate film cooling

Each of these mechanisms is treated separately and along pre-selected sections of the exchanger. The procedure for determining the last two of the mechanisms follows that described earlier for single phase heat transfer. The following expression is used to calculate the film coefficient for the condensing vapor:

$$h_c = \frac{8.33 \times 10^3}{(M_c/L_c \cdot N_s)^{.33}} \times k_f \times \left[\frac{S g_c^2}{\mu_f}\right]^{0.33}$$

where

h_c = condensing film coefficient.
M_c = mass condensed in lbs/hr
L_c = tube length for condensation.
$= \frac{A_{zone}}{A} \times (L - 0.5)$

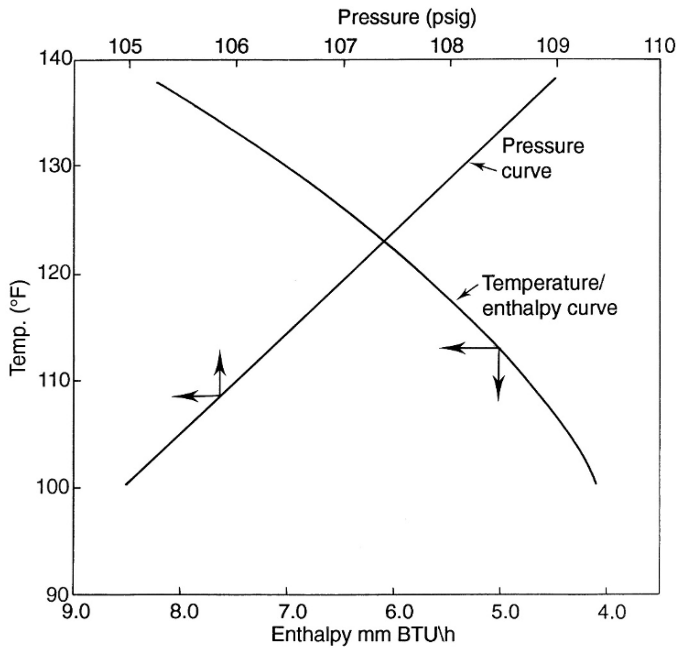


Figure 18.43. Enthalpy curve for a de-butanizer overhead condenser.

$N_s = 2.08\ N_t^{0.495}$ for triangular pitch.
k_f = thermal conductivity of condensate at film temperature.
Sg = specific gravity of condensate.
μ_f = viscosity of condensate at film temperature in Cps

Again there are many excellent computer programs that calculate condenser thermal ratings, and these of course save the tedium of the manual calculation. However no matter which method of calculation is selected there is required one major additional piece of data over that required for single phase heat exchange. That item is the enthalpy curve for the vapor.

Enthalpy curves are given as the heat content per lb or per hour contained in the mixed phase condensing fluid plotted against temperature. An example of such a curve is given in Figure 18.43. These Enthalpy curves are developed from the vapor/liquid or flash calculations described in Chapter 3 of this volume.

Briefly the calculation for the curve commences with determining the dew point of the vapor and the bubble point of the condensate. Three or more temperatures are

selected between the dew and bubble points and the V/L calculation of the fluid at these temperatures carried out. Enthalpy for the vapor phase and the liquid phase are added for each composition of the phases at the selected temperatures. These together with the enthalpy at dew point and bubble point are then plotted.

As in the case of the shell and tube exchanger and the air cooler a manual calculation for condensers is described here. Again this is done to provide some understanding of the data required to size such a unit and its significance in the calculation procedure. Computer aided designs should however be used for these calculations whenever possible.

The following calculation steps describe a method for calculating the film coefficient of a vapor condensing on the shell side of a S&T exchanger. The complete rating calculation will not be given here as much of the remaining calculation is simply repetitive.

Step 1. Calculate the dew point of the vapor stream at its source pressure. Estimate the pressure drop across the system. Usually 3–5 psi will account for piping and the exchanger pressure drop. Calculate the bubble point of the condensate at the terminal pressure. Select three or more temperatures between dew point and bubble point and calculate the vapor/liquid quantities at these conditions of temperature and pressure.

Step 2. Calculate the enthalpy of the vapor and liquid at these temperatures. Plot the total enthalpies against temperature to construct the enthalpy curve. Establish the properties of the vapor phase and liquid phase for each temperature interval. The properties mostly required are Sg, viscosity, Mole wt, thermal conductivity, and specific heats.

Step 3. In the case of a water cooler calculate the duty of the exchanger and the quantity of water in lbs/hr. Commence the heat transfer calculation by assuming an overall heat transfer coefficient (use the data given in Table A9.1). Calculating the corrected LMTD, and the surface area.

Step 4. Using the surface area calculated in step 3 define the exchanger geometry in terms of number of tube passes, number of tubes on the center line, shell diameter, baffle arrangement and the shell free flow area. Calculate also the water flow in feet per sec.

Step 5. Divide the exchanger into 3 or 4 zones by selecting the zone temperatures on the enthalpy curve. Calculate the average weight of vapor and the average weight of condensate in each zone. Using these averages calculate the average heat transferred for:

Cooling of the vapor Q_{v}

Cooling of the condensate Q_{L}

Condensing of the vapor which will be:

Total heat in the zone (from the enthalpy curve) less the sum of Q_{v} and Q_{L}.

Step 6. Calculate the film coefficient for the tube side fluid. See previous item 'Estimating Shell and Tube Surface Area and Pressure Drop'.

Step 7. Starting with zone 1 and knowing the outlet temperature of the coolant fluid, the total heat duty of the zone, and the shell side temperatures calculate the coolant inlet temperature. Using this calculate the LMTD for the zone and, assuming a zone overall heat transfer coefficient U, calculate a surface area for the zone. Using this and the total exchanger area estimated in step 4 establish L_c in feet.

Step 8. Calculate the condensing film coefficient from the equation given earlier. This will be an uncorrected value for h_c. This will be corrected to account for turbulence by the expression:

$$h_{c\,(corr)} = h_c \times (G_v/5)$$

where

$$G_v = \text{average vapor mass velocity in lbs/hr} \cdot \text{sqft}$$

Step 9. Calculate the value of G_v using the free flow area allocated to the vapor γ_v. The following expressions are used for this:

$$\gamma_v = 1 - \gamma_L$$

$$\frac{1}{\gamma_L} = 1 + \frac{\text{Ave mass vapor}}{\text{Ave mass liquid}} \times (\mu_v/\mu_L)^{0.111} \times (\rho_L/\rho_v)^{0.555}$$

$$G_v = \frac{\text{Ave mass Vapor}}{25 \times \text{Free flow area} \times \gamma_v}$$

Step 10. Calculate the film coefficient hv for the vapor cooling mechanism. This will be the procedure used for a single phase cooling given in a previous item. This is corrected to account for resistance of the condensate film by the expression:

$$\frac{1}{h_{v\,corr}} = -\frac{1}{h_c} + \frac{1}{1.25 h_v}$$

Step 11. Calculate the film coefficient for the condensate cooling mechanism. Again this is the procedure described in the Item for single phase cooling on the shell side. This is corrected for drip cooling that occurs over a tube bank.

$$\text{Drip cooling}\, h_{dc} = 1.5 \times h_c$$

$$\text{and}\, h_L \text{ corrected} = \frac{2 \times h_{dc} \times h_L}{h_{dc} + h_L}$$

Step 12. Calculate the total zone film coefficient ho using the following expression:

$$h_o = -\frac{Q_{zone}}{\frac{Q_c}{h_c} + \frac{Q_v}{h_v} + \frac{Q_L}{h_L}}$$

where
Q_c, Q_v, Q_L are the enthalpies for condensing, vapor cooling, and condensate cooling, respectively.

Step 13. Calculate the overall heat transfer coefficient neglecting the shell side coefficient from Step 12. Thus:

$$\frac{1}{U_x} = r_o + r_w + r_{io} + R_{io}$$

where
r_o, r_w, r_{io} are fouling factors for shell fluid, wall, and tube side fluid respectively.
R_{io} is the tube side film coefficient calculated in Step 6.

Step 14. Calculate the overall heat transfer coefficient U_{zone} for the zone using the expression:

$$U_{zone} = \frac{h_o \times U_x}{h_o + U_x}$$

Check the calculated U against that assumed for the zone. Repeat the calculation if necessary to make a match.

Step 15. Calculate the zone area using the acceptable calculated U. Repeat steps 7 through 14 for the other zones. The total surface area is the sum of those for each zone.

Reboilers

Reboilers are used in fractionation to provide a heat source to the system, and to generate a stripping vapor stream to the tower. Reboilers are operated either by the natural circulation of a fluid or by forced circulation of the fluid to be reboiled. This chapter deals only with natural circulation reboilers.

There are three common types of reboilers and these are:

- The kettle type reboiler
- The once through thermosyphon reboiler
- The re-circulating thermosyphon reboiler

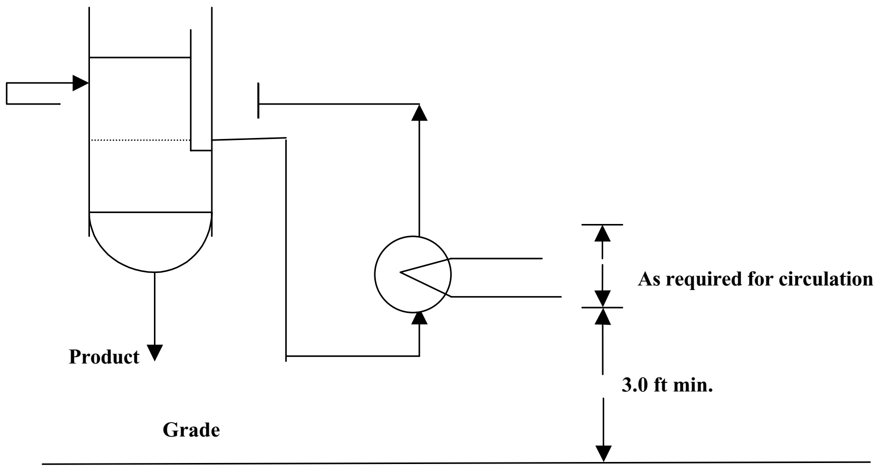
The kettle reboiler

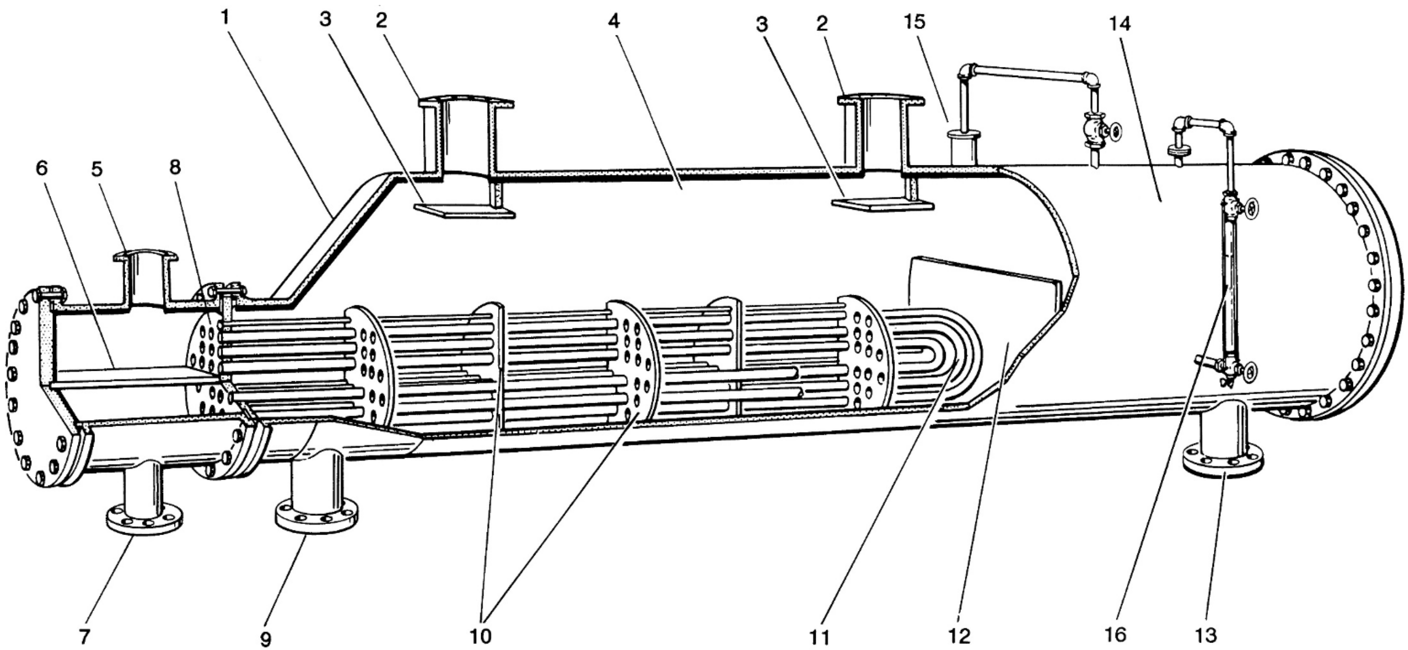
This type of reboiler (Figure 18.44) is extremely versatile. It can handle a very wide range of vaporization loads. (e.g., when used as LPG vaporizer for fuel gas purposes it vaporizes 100% of the feed). The equipment consists of a large shell into which is fitted a tube bundle through which the heating medium flows. The liquid to be reboiled enters the bottom of the shell at the end adjacent to the tube inlet/outlet chamber. The liquid is boiled and partially vaporized by flowing across the tube bundle. The diameter of the shell is sized such that there is sufficient space above the tube bundle and the top of the shell to allow some disengaging of the liquid and vapor. A baffle weir is installed at the end of the tube bundle furthest from the inlet. This baffle weir establishes a liquid level over the tube bundle in the shell. The boiling liquid flows over this weir to the shell outlet nozzle, while the vapor generated is allowed to exit from the top of the shell through one or two nozzles.

The space downstream of the weir is sized for liquid holdup to satisfy the surge requirements for the product. Thus it is not necessary to provide space in the bottom of the tower for product surge. If the heating medium is non fouling it is permissible to use U tubes for the tube bundle. Otherwise the tube bundle must be of the floating head type. The kettle type reboiler should always be the first type to be considered if there are no elevation constraints to pumping the bottoms product away.

Once through thermosyphon reboiler

This type of reboiler and its location relative to the tower is given in the sketch below:





1 shell ; 2 Shell outlet nozzles Vapor ; 3. Entrainment Baffles ; 4 Vapor Disengaging Space. 5. Channel Inlet Nozzle; 6. Channel Partition. ; 7.Channel outlet nozzle. ; 8. Tube Sheet; 9. Shell inlet nozzle.; 10. Tube support sheets.; 11 U Tube returns.; 12 Weir. ; 13 Shell outlet nozzle (liquid),; 14. Liquid hold up (Surge) section.; 15Top of level – instrument housing (external displacer).; 16. Liquid level gauge.

Figure 18.44. The components of a kettle reboiler.

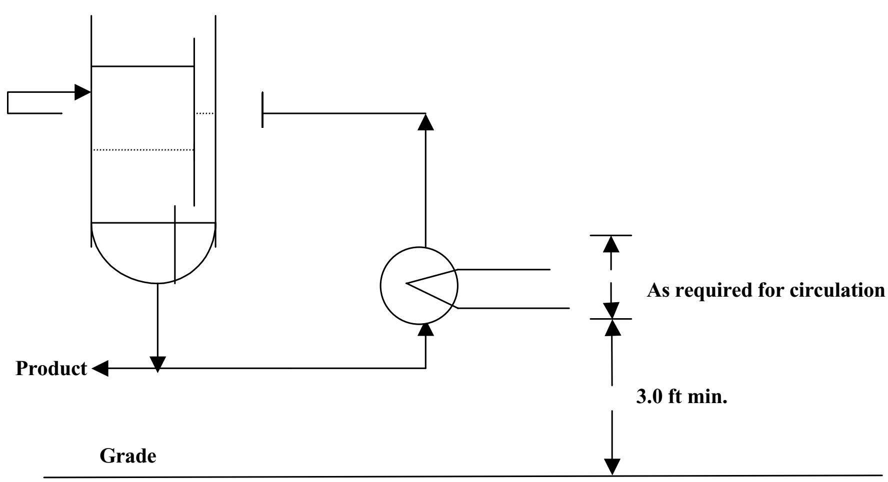
This type of reboiler should be considered when a relatively high amount of surge is required for the bottom product and when it is necessary to provide head for the product pump (NPSH requirement).

This type of reboiler takes the liquid from the bottom tray of the fractionator as feed. This stream enters the shell side of a vertical single tube pass shell and tube exchanger by gravity head to the bottom of the shell. The heating medium flows tube side to partially vaporize the liquid feed. A siphoning effect is caused by the difference in density between the reboiler feed and the vapor/liquid effluent. This allows the reboiler effluent to exit from the top of the shell side and reenter the tower where the vapor disengages from the liquid phase. The liquid is the bottom product of the fractionator and is discharged from the bottom of the tower.

Both the kettle type and the once through thermo-siphon type constitute a theoretical tray as regards fractionation. Unlike the kettle reboiler the once through thermo-siphon is limited to a vaporization of not more than 60% of the feed. The low holdup of the feed from a tray results in severe surging through the reboiler at high vaporization rates.

The re-circulating thermo-siphon reboiler

When vaporization rates higher than 60% of reboiler feed is required and a kettle re-boiler is unsuitable a recirculating Thermo-siphon type reboiler should be considered. A sketch showing this type of reboiler is given below:



This reboiler is similar to the once through thermo-siphon in as far as it operates by flowing a liquid feed through the shell side of the vertical reboiler by the siphon

mechanism. In the case of the recirculating reboiler however the feed to the reboiler is a stream of the bottom product from the fractionator. This is vaporized as described earlier and the liquid/vapor effluent returned to the tower. The vaporization by this reboiler can exceed 60% without danger of surging. However vaporization in this type of reboiler should not exceed 80%. Its action is directed solely to imputing heat to the tower and, because it recycles the same composition stream to the tower bottom, it cannot be considered as a theoretical fractionating tray. (Although some amount of fractionation does occur in this system).

Note in the description of both the thermo-siphon type reboilers the heating fluid is shown as flowing tube side. There may be cases where this stream will be routed shell side and the reboil fluid directed tube side. Some guidance to this selection is provided by the following preference for tube side fluid:

1. Corrosive or fouling fluids.
2. The less viscous of the two fluids.
3. The fluid under the higher pressure.
4. Condensing steam.

Reboiler sizing

As in the case of most heat exchangers the sizing calculation is quite rigorous and complex. Normally process engineers rarely need to compute this in detail. There will be need however to estimate the size of these items for cost purposes or for plot layout studies. This sizing is greatly simplified by applying heat flux quantities to the predetermined reboiler duty. Heat flux is the value of heat transferred per unit time per sqft of surface. The following list gives a range of heat fluxes that have been used in design and observed in operating units.

	Design	Observed
	(Btu/hr · sqft)	
Kettle type	12,000	15,000–20,000
Once through	15,000	17,500+
Recirculating	15,000	up to 20,000
Forced circulation	20,000	–

The duty of the reboiler is obtained by the overall heat balance over the tower. This is accomplished by equating the total heat out of a fractionating tower to the heat supplied, making the reboiler duty the unknown in the heat supplied statement. Now the heat out of the fractionator is the total heat in the products leaving plus the condenser duty. The heat supplied to the tower is the heat brought in with the feed, and the heat supplied by the reboiler.

Example calculation

The feed to a fractionator is 87,960 lbs/hr of mixed hydrocarbons. It enters the tower as a vapor and liquid stream and has a total enthalpy of 15.134 mm Btu/hr. The overhead products are a distillate and a vapor stream at 95°F. The vapor is 1,590 lb/hr with an enthalpy of 320 Btu/lb. The distillate is 8,028 lbs/hr with an enthalpy of 170 Btu/hr. The bottom product from the tower is 78,342 lbs/hr and leaves as a liquid at its boiling point at 440°F. Its enthalpy is 370 Btu/lb. The overhead condenser duty is 4.278 mm Btu/lb. Calculate the reboiler duty.

Calculation

Calculate the reboiler duty from the overall tower heat balance as in the following:

	V/L	°API	°F	lbs/hr	Enthalpy, Btu/lb	Enthalpy, mm Btu/hr
In						
Feed	VL	–	300	87,960	–	15.134
Reboiler duty						x
Total In				87,960		$15.134 + x$
Out						
Bottom Prod	L	–	440	78,342	370	28.986
O/head Dist	L	–	95	8,028	170	1.364
O/head vapor	V	–	95	1,590	320	0.508
Condenser duty						4.278
Total out				87,960		35.136

$$\text{Heat In} = \text{Heat Out}$$

Then

$$\begin{aligned} 15.134 + x &= 35.136 \\ \text{Reboiler duty}\, x &= 20.002\,\text{mm Btu/hr.} \end{aligned}$$

Using a heat flux of 15,000 Btu/hr sqft the surface area for the reboiler becomes

$$\frac{20002000}{15{,}000} = 1{,}333.5\,\text{sqft.}$$

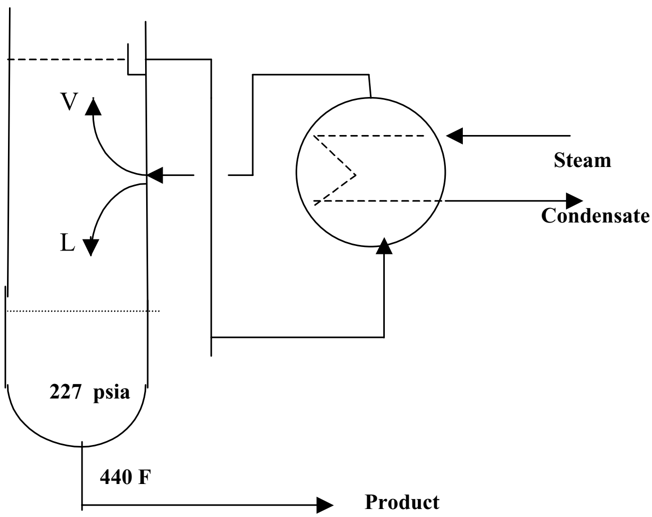
Estimating the liquid and vapor flow from the reboiler

It is necessary to know the vapor and liquid flow leaving the reboiler and entering the tower for the following reasons:

- To establish that there is sufficient vapor rising in the tower to strip the bottom product effectively
- To establish the vapor loading to the bottom tray for calculating the tray loading

- To be able to calculate the driving force for flow through the exchanger in the case of thermo-siphon reboilers

The calculation of this flow is again based on a heat balance. In this case it is the heat balance across the reboiler itself. With the duty of the reboiler now established by the overall tower heat balance, as described above, the balance over the reboiler can proceed as follows:



	V/L	°API	°F	lbs/hr	Enthalpy, Btu/lb mm Btu/hr
In					
Liquid from tray 1	L	–	430	78,342 + V	369 28.908 + 369 V
Reboiler duty					20.002
Total In				7,834 + V	48.910 + 369 V
Out					
Bottom prod	L	–	440	78,342	370 + 28.986
Vapor to tray 1	V	–	440	V	458 + 458 V
Total Out				78,342 + V	28.986 + 458 V

The temperature of the bottom tray (430°F) is estimated from a straight line temperature profile of the tower. As a rule of thumb—for a 30 to 40 tray tower the bottom tray will be about 10°F lower than the bottom temperature.

$$\text{Again Heat In} = \text{Heat Out}$$

Then $$48.910 + 369\text{ V} = 28.986 + 458\text{ V}$$

$$\text{V} = 223{,}865\text{ lbs/hr}$$

Now the mole weight of the vapor is determined from the bubble point calculation of the bottom product used to determine the tower bottom temperature (see Chapter 1). In the case of the calculation example given above the bubble point calculation for the bottom product was as follows:

$$\text{Pressure at bottom of tower} = 220 + (30 \times 0.25)$$
$$= 227\,\text{psia}$$

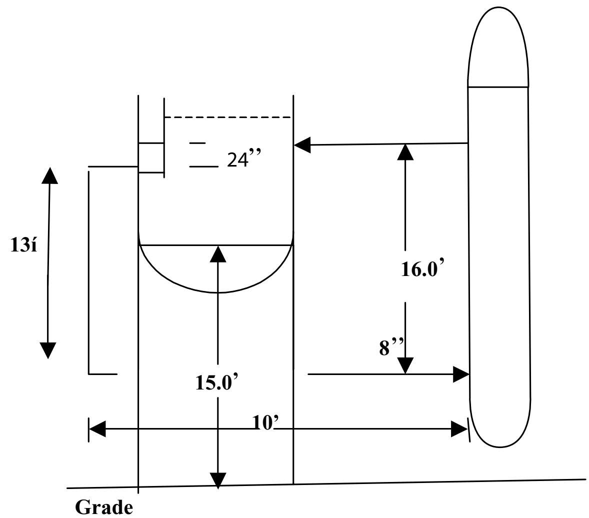
	X_W	1st Trial = 400°F K	1st Trial = 400°F $Y = XK$	2nd Trial = 435°F K	2nd Trial = 435°F $Y = XK$	MW	Weight factor	lbs/gal	Vol factor
nC_4	0.017	2.7	0.046	3.1	0.053	58	3.1	4.86	0.64
iC_5	0.047	1.9	0.089	2.2	0.103	72	7.4	5.20	1.42 (°API 53.1)
nC_5	0.055	1.7	0.094	1.9	0.105	72	7.6	5.25	1.45
C_6	0.345	0.96	0.331	1.3	0.449	81	36.4	6.83	5.33
C_7	0.322	0.44	0.142	0.59	0.190	102	19.4	6.84	2.84
C_8	0.214	0.17	0.036	0.25	0.054	128	6.9	6.94	0.99
Total	1.000		0.738		0.954	84.7	80.8	6.38	12.67

Actual temp = 440°F

The mole weight of the vapor is that calculated for the "y" column in the above table which is 84.7.

Calculating the pressure head driving force through a thermo-siphon reboiler

The big advantage of thermo-siphon reboilers is that there are no working parts, such as pumps, that can go wrong and cause failure. However a major cause of failure in a thermo-siphon reboiler is loss of driving force to move the fluid over or through the tube bundle. This problem mostly occurs during commissioning when the reboiler has been incorrectly positioned relative to the tower nozzles, or during start up where debris left after maintenance blocks one or other of the nozzles. In both these cases the problem is really the loss of pressure head that drives the fluid to be reboiled through the exchanger. The calculation to determine the theoretical driving force is based on the density of the incoming liquid, the head of that liquid to the inlet nozzle, the density of the out flowing liquid/vapor fluid, and its head. An example of a pressure driving force calculation based on a once through thermo-siphon (as shown in the diagram below) is as follows. The flow data is based on the heat balance given earlier in this item.



The density of the liquid to the reboiler is 38.2 lbs/cuft at 430°F and the total flow is 302,207 lbs/hr.

$$\text{Hot cuft/hr} = 7,911$$
$$\text{Hot gpm} = 966$$

The transfer line from the tower to the bottom nozzle of the reboiler is a 8″ schedule 40 seamless steel pipe. The head between the bottom of the tower draw off pot and the reboiler nozzle is 13 ft. The equivalent horizontal line length including fittings is 15 ft.

From the friction loss tables in the Appendix, head loss due to friction = 66.4 ft/100 ft of line (viscosity is taken as 1.1 cs).

$$\begin{aligned}
\text{Total line length to the reboiler is } 13 + 15 \text{ ft} &= 28 \text{ ft.} \\
\text{Head loss due to friction} = \frac{28 \times 66.4}{1,000} &= 1.85 \text{ ft.} \\
\text{Head of liquid in draw off pot} &= 24'' \\
\text{Head of liquid to the reboiler inlet nozzle} = 13 + 2 \text{ ft.} &= 15 \text{ ft.} \\
\text{Pressure head at the reboiler inlet} = 15 - 1.85 \text{ ft} &= 13.15 \text{ ft} \\
&= \frac{13.15 \times 38.2}{144} \\
&= 3.49 \text{ psi}
\end{aligned}$$

The density of the vapor/liquid stream leaving the reboiler is calculated as follows:

$$\frac{\text{Total mass of fluid}}{\text{cuft liquid} + \text{cuft vapor}}.$$

lbs/cuft of liquid (this is bottom product) = 39.4 @ 440°F
mole wt of vapor (see bubble point calculation above) = 84.74

$$\text{cuft/hr of liquid} = \frac{78{,}342 \text{ lbs/hr}}{39.9} = 1{,}963.5$$

$$\begin{aligned}\text{cuft/hr of vapor} &= \frac{223{,}865}{84.5}\text{lbs/hr} = 2{,}643 \text{ moles/hr}\\ &= \frac{2{,}643 \times 378 \times 14.7 \times (460 + 440°\text{F})}{227 \times 520}\\ &= 111{,}974.6 \text{ cuft/hr}\end{aligned}$$

$$\begin{aligned}\text{density of fluid from reboiler} &= \frac{302{,}207}{1{,}963.5 + 111{,}974.6}\\ &= 2.65 \text{ lbs/cuft.}\end{aligned}$$

In this case the fluid to be reboiled flows on the shell side of the exchanger. The manufacturer's certified shell side pressure drop based on all vapor flow is 1.5 psi. The mixed phase pressure drop is calculated using Figure 18.46 thus.

$$m_v = \text{average mass of vapor} = \frac{223{,}865}{2} = 111{,}933 \text{ lbs/hr}$$

$$m_l = \text{average mass of liquid} = \frac{302{,}207 + 78{,}342}{2} = 190{,}275 \text{ lbs/hr}$$

$$\rho_v = \text{average density of vapor} = \frac{2.0}{2} = 1.0 \text{ lbs/cuft.}$$

$$\rho_l = \text{average density of liquid} = \frac{38.8 + 39.9}{2} = 39.35 \text{ lbs/cuft}$$

Referring to Figure 18.45

$$\begin{aligned}R_m &= \frac{1}{\dfrac{m_v}{m_l} + \dfrac{\rho_v}{\rho_l}}\\ &= \frac{1}{\dfrac{111{,}933}{190{,}275} + \dfrac{1.0}{39.32}}\\ &= 1.63\end{aligned}$$

From Figure 18.45

$$\alpha = 0.42$$

$$\begin{aligned}\Delta P_{\text{mixed phase}} &= \alpha \times \Delta P_{\text{gas}}\\ &= 0.42 \times 1.5 \text{ psi}\\ &= 0.63 \text{ psi.}\end{aligned}$$

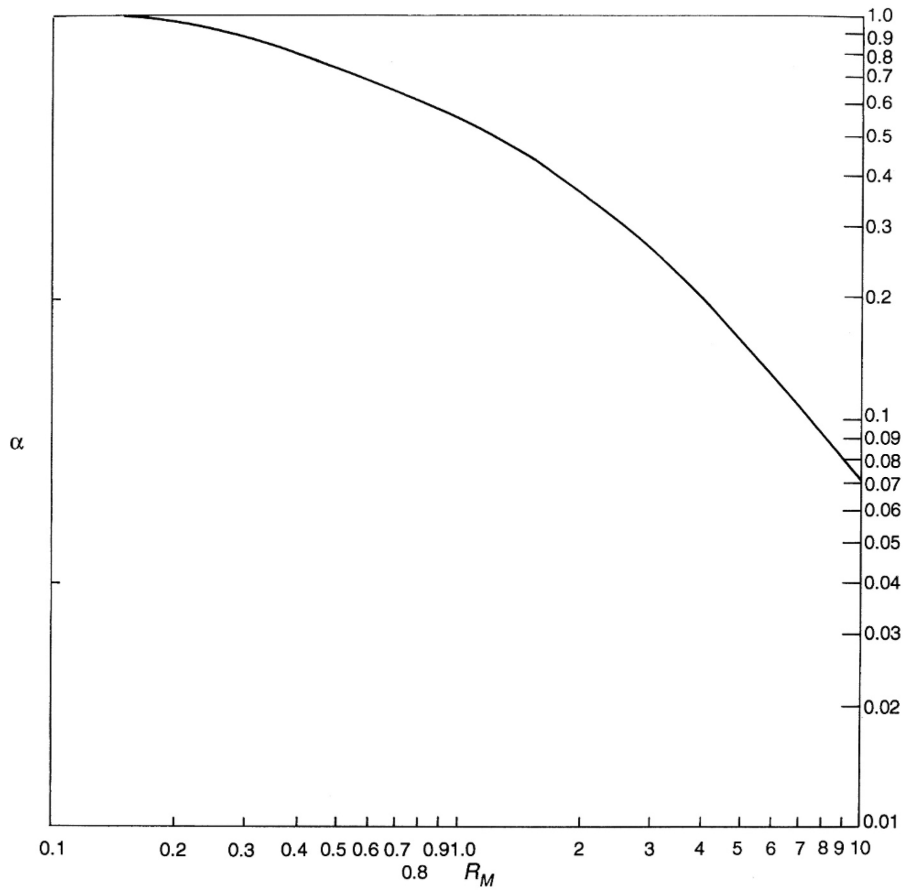


Figure 18.45. Two phase pressure drop factor for flow across staggered tubes.

To calculate the total head of liquid from exchanger inlet to outlet nozzle.

Tube length is standard 16 ft.

Assume bottom 20% of length is all liquid phase at a density of 38.8 lbs/cuft.

Then head is $\dfrac{16 \times 0.2 \times 38.8}{144} = 0.862$ psi

Remaining head is mixed phase at a density of 2.65 lbs/cuft.

Then head is $\dfrac{16 \times 0.8 \times 2.65}{144} = 0.24$ psi

Neglecting the small pressure drop due to friction in the 4 ft long return line to tower,

$$\begin{aligned} \text{Friction loss through exchanger} &= 0.630 \text{ psi} \\ \text{Lower section head} &= 0.862 \text{ psi} \\ \text{Upper section head} &= 0.240 \text{ psi} \\ \text{TOTAL} &= 1.732 \text{ psi} \end{aligned}$$

$$\begin{aligned} \text{Then driving force} &= \text{Pressure head available – pressure head required.} \\ &= 3.49\,\text{psi} - 1.732\text{psi} \\ &= 1.758\,\text{psi which is satisfactory for good flow.} \end{aligned}$$

Note the height of the tower above grade is usually fixed by pump suction requirements in the first place. It may be adjusted upwards if necessary to accommodate a head to a reboiler. However this necessity is quite rare. The transfer line to the reboiler should have its horizontal section at least 3.0 ft above grade to allow for maintenance, etc.

18.5 Fired Heaters

Types of fired heaters

This chapter provides some features and detail of fired heaters.

Most chemical plants and all petroleum refineries contain fired heaters as a means of providing heat energy into a system. Because the equipment utilizes an outside source of fuel it is usually supported and enhanced by a heat exchange system to minimize the quantity of fuel required.

Generally fired heaters fall into two major categories:

- Horizontal type
- Vertical type

The horizontal type heater usually means a box type heater with the tubes running horizontally along the walls. Vertical type is normally a cylindrical heater containing vertical tubes. Figures 18.46 and 18.47 show examples of these two types of heaters.

These figures also give some nomenclature used in describing these items of equipment. Other terms used in connection with fired heaters are as follows:

Headers & Return Bends: are the fittings used to connect individual tubes.

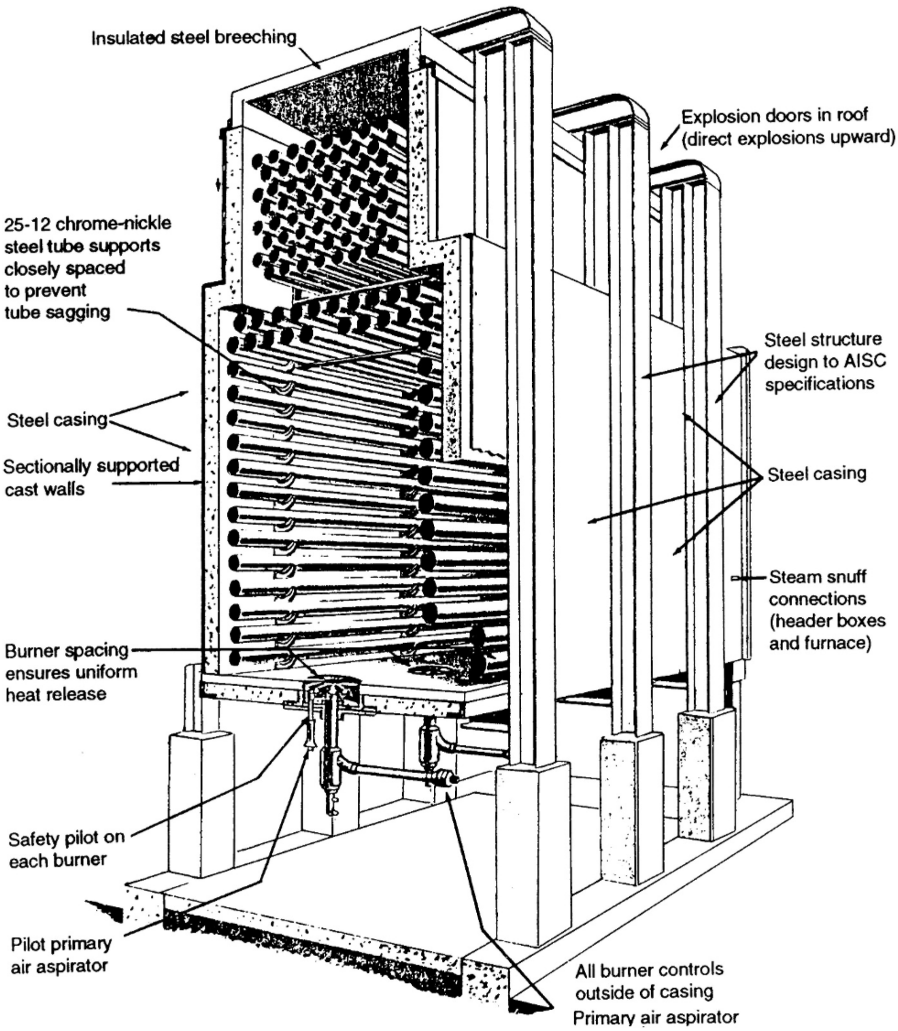


Figure 18.46. Horizontal type heater.

Terminals: are the inlet and outlet connections.

Crossovers: are the piping used to connect the radiant with the convection section; usually external to the heater.

Manifold: is the external piping used to connect the heater passes to the process piping; may be furnished with the heater.

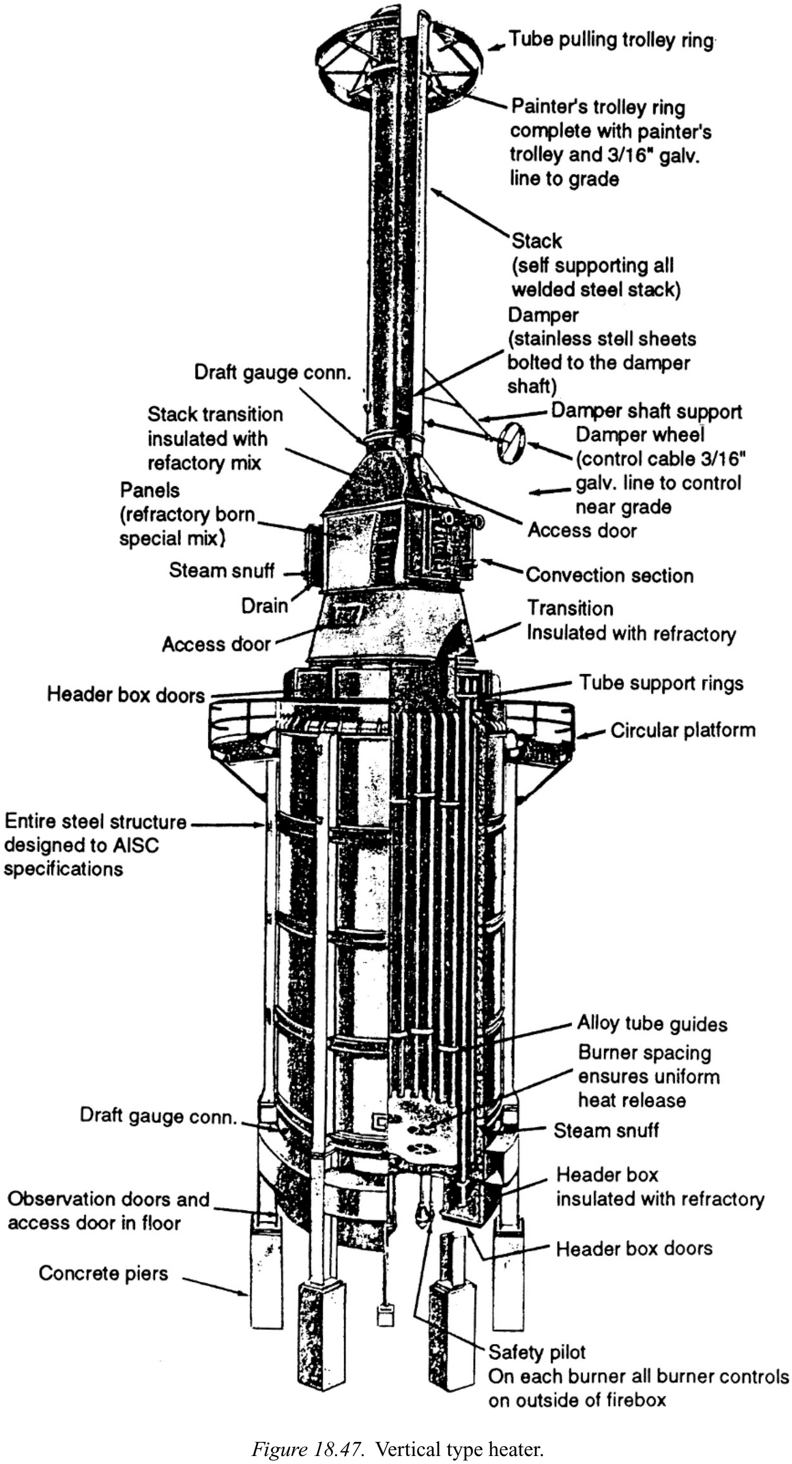


Figure 18.47. Vertical type heater.

Setting: any and all parts that form:

(a) coil supports
(b) enclosure (housing)

Casing: Is the steel shell which encloses the heater.

Bridge wall or partition wall: are the refractory walls inside the heater that divide the radiant section into separately fired zones.

Shield tubes or shock tubes: are the first 2 or 3 rows of tubes in the convection section. They protect or shield the convection tubes from direct radiant heat and must have the same metallurgy as the radiant tubes and have no fins.

Air plenum: is the chamber enclosing burners under the heater and having louvers to control the air flow.

Cylindrical heaters require less plot space and are usually less expensive. They also have better radiant symmetry than the horizontal type.

Horizontal box types are preferred for crude oil heaters, although vertical cylindrical heaters have been used in this service. Vacuum unit heaters should have horizontal tubes to eliminate the static head pressure at the bottom of vertical tubes and to reduce the possibility of two-phase slugging in the large exit tubes.

Occasionally, several different services ("coils") may be placed in a single heater with a cost saving. This is possible if the services are closely tied to each other in the process. Catalytic reforming pre-heater and re-heaters in one casing is an example. Reactor heater and stripper reboiler in one casing is another example. This arrangement is made possible by using a refractory partition wall to separate the radiant coils. The separate radiant coils may be controlled separately over a wide range of conditions by means of their own controls and burners. If a convection section is used, it is usually common to the several services. If maintenance on one coil is required, the entire heater must be shut down. Also, the range of controllability is less than with separate heaters.

Each of these types may be shop fabricated if size permits. Shop fabrication reduces costs. However, shop fabrication should not be forced to the extent of getting an improperly proportioned heater.

Codes and standards

Fired heaters have a "live" source of energy. That is, they use a flammable material in order to impart heat energy to a process stream. Because of this the design,

construction and operation of process fired heaters and boilers are strictly controlled by legislative and other codes and standards. This item outlines some of the more important of these codes and standards which need to be recognized by engineers dealing with fired heaters in any way.

Codes and standards directly applicable to fired heaters are listed below. In addition, there are many codes and standards covering such factors as materials, welding, refractories, structural steel, etc., which apply to fired heaters.

API RP-530

Calculation of heater tube thickness' in refineries

This recommended practice sets forth procedures for calculating the wall thickness of heater tubes for service at elevated temperatures in petroleum refineries.

API Standard 630

Tube and heater dimensions for fired heaters for refinery service

This standard establishes certain standard dimensions for heater tubes and for cast and wrought headers.

Tube sizes and header centre-to-centre dimensions covered by the standard are as follows:

Tube OD, inches		Header C-TO-C, inches	
Primary	Secondary	Group A	Group B
2.375	–	4.00	4.75
2.875	–	5.00	5.25
3.50	–	6.00	–
4.00	–	7.00	6.50
4.50	–	8.00	7.25
–	5.00	9.00	7.75
5.563	–	10.00	8.50
–	6.00	11.00	9.00
6.625	–	12.00	10.00
–	7.625	14.00	12.00
8.625	–	16.00	14.00

Groove dimensions and tolerances for rolled headers are also given. Much of this standard is also used in chemical and petrochemical plants.

API RP-2002 Fire protection in natural gasoline plants

This practice contains a brief statement about the use of snuffing steam. This system provides the piping of a steam source to the heater fire box which in an emergency can introduce steam into the box to quench any uncontrolled fire. This system may be automatically controlled or activated manually.

API Guide for Inspection of Refinery Equipment, Chapter IX, Fired Heaters and Stacks.

This reference gives a general description of fired heaters and describes how to inspect them, what damage to look for, and how to report the results of the inspection.

ASME boiler code and boiler codes of the USA

These are applicable in process plants if steam is generated, superheated or boiler feed water preheated in the convection section. Special materials are required according to ASME Section I. The external piping and pressure relieving devices must also be in accordance with ASME Section I.

Contractors' standards

These will be covered in the "Narrative Specification" for the particular job and/or heater. The narrative specification is written by the heat transfer engineer specialists in the Contractors' Mechanical Equipment Group. These specifications detail all of the pertinent aspects required in the manufacture of the equipment. It will encompass all of the requirements of the applicable.

Thermal rating

Refinery process engineers are seldom if ever required to thermal rate a fired heater or indeed check the thermal rating. This is a procedure that falls in the realm of specialist mechanical engineers with extensive experience in heater design and fabrication. Process engineers are however required to specify the equipment so that it can be designed and installed to meet the requirements of the process heat balance. To do this effectively it is desirable to know something about the mechanism of heater thermal rating.

A fired heater is essentially a heat exchanger in which most of the heat is transferred by radiation instead of by convection and conduction. Rating involves a heat balance between the heat releasing and heat absorbing streams, and a rate relationship.

Fuel is burned in a combustion chamber to produce a “flame burst”. The theoretical flame burst temperature may vary from 4,000°F when burning refinery gases with 20% excess air preheated to 460°F, down to 2,300°F when burning residual fuel oils with 100% excess air at 60°F. Heat is transferred from the flame burst to the gases in the firebox by radiation and mixing of the products of combustion. Heat is then transferred from the firebox gases to the tubes mainly by radiation.

The common practice is to assume a single temperature for the firebox gases for the purpose of radiation calculations. This temperature may be the same as the exit gas temperature from the firebox to the convection section (bridge wall temperature), or it may be different due to the shape of the heater and to the effect of convection heat transfer in the radiant section. Experience with the particular type of heater is required in order to select the effective firebox temperature accurately.

Whilst this chapter does not detail the rating procedure or give an example calculation the following steps summarize the rating procedure:

1.0 Calculate net heat release and fuel quantity burned from the specified heat absorption duty and an assumed or specified efficiency.
2.0 Select excess air percentage and determine flue gas rates.
3.0 Calculate duty in the radiant section by assuming 70% of total duty is radiant. This is a typical figure and will be checked later in the calculations. For very high process temperatures, such as in steam-methane reforming heaters, the radiant duty may be as low as 45% of the total.
4.0 Calculate the average process fluid temperature in the radiant section and add 100°F to get the tube wall temperature. The figure of 100°F is usually a good first guess and can be checked later by using the calculated inside film coefficient and metal resistance.
5.0 Calculate the radiant surface area using the average allowable flux. Convection surface is usually about equal to the radiant surface.
6.0 Select a tube size and pass arrangement that will give the required total surface and meet specified pressure drop limitations.
7.0 Select a center-to-center spacing for the tubes from the API 630 Standard or from dimensions of standard fittings, and calculate firebox dimensions. Long furnaces minimize the number of return bends and thus reduce cost. Shorter and wider fireboxes usually give more uniform heat distribution and lessen the probability of flame impingement on the tubes. For vertical cylindrical heaters, the ratio of radiant tube length to tube circle diameter should not exceed 2.7.
8.0 The remainder of the calculation involves determining the firebox exit temperature from assumption (3) above, applying an experience factor for the type of furnace to obtain the average firebox temperature, and then checking if this temperature will transfer the required radiant heat.

9.0 The average heat flux (proportional to radiant surface) and the percent of total duty of the radiant section (which affects average tube wall temperature) are varied until a balance is obtained. The convection section surface and arrangement can now be calculated.

10.0 The heater is normally designed to allow adequate draft at the burners with at least 125% of design heat release and an additional 10% excess air at the maximum and minimum ambient air temperatures.

Heat flux

Although a process engineer is not normally required to thermally rate a process heater he is often required to estimate the heater size, for preliminary cost estimates or plot layout and the like. This can be accomplished quite simply by the use of "Heat Flux". Whilst this figure is quoted as Btu/hr sqft, it is not however an overall heat transfer coefficient, it lacks the driving force ΔT for this. Heat flux is the rate of heat transmission through the tubes into the process fluid.

The maximum film temperature and tube metal temperature are a function of heat flux and the inside film heat transfer coefficient.

The heat flux varies around the circumference of the tube, being a maximum on the side facing the firebox. The value depends upon the sum of the heat received directly from the firebox radiation and the heat re-radiated from the refractory.

Single fired process heaters are usually specified for a maximum average heat flux of 10,000–12,000 Btu/hr sqft. The maximum point heat flux is about 1.8 times greater.

Double fired heaters are usually specified for about $13{,}500 \times 18{,}000$ Btu/hr sqft *average* heat flux with the maximum point flux being about 1.2 times greater.

The following are typical flux values for heaters in hydrocarbon service:

Horizontal, fired on one side	8,000–12,000
Vertical, fired on one side from bottom	9,000–12,000
Vertical, single row, fired on both sides	13,000–18,000

Heater efficiency

The efficiency of a fired heater is the ratio of the heat absorbed by the process fluid to the heat released by combustion of the fuel expressed as a percentage.

Heat release may be based on the lower heating value (LHV) of the fuel or higher heating value (HHV). Process heaters are usually based on LHV and boilers on HHV. The HHV efficiency is lower than the LHV efficiency by the ratio of the two heating values.

Heat is wasted from a fired heater in two ways:

- with the hot stack gas,
- by radiation and convection from the setting.

The major loss is by the heat contained in the stack gas. The temperature of the stack gas is determined by the temperature of the incoming process fluid unless an air preheater is used. The closest economical approach to process fluid is about 100°F. If the major process stream is very hot at the inlet, it may be possible to find a colder process stream to pass through the convection section to improve efficiency, provided plant control and flexibility are adequately provided for. A more common method of improving efficiency is to generate and/or superheat steam and preheat boiler feed water.

The lowest stack temperature that can be used is determined by the dew point of the stack gases. See the section on stack emissions.

Figures 18.48 and 18.49 may be used to estimate flue gas heat loss.

The loss to flue gas is expressed as a percentage of the total heat of combustion available from the fuel. These figures also show the effect of excess air on efficiency. Typically excess air for efficiency guarantees is 20% when firing fuel gas and 30% when firing oil.

Heat loss from the setting, called radiation loss, is about 1½ to 2% of the heat release.

The range of efficiencies is approximately as follows:

Very high	—	90%+. Large boilers and process heaters with air pre-heaters.
High	—	85%. Large heaters with low process inlet temperatures and/or air pre-heaters.
Usual	—	70–80%.
Low	—	60% and less. All radiant.

Engineers are often required to check the efficiencies of the process heaters on operating units assigned to them. This can be done using the heats of combustion given in Figures 18.A.2 and 18.A.3 in the Appendix to this Chapter. The steps used to carry out these calculations are as follows:

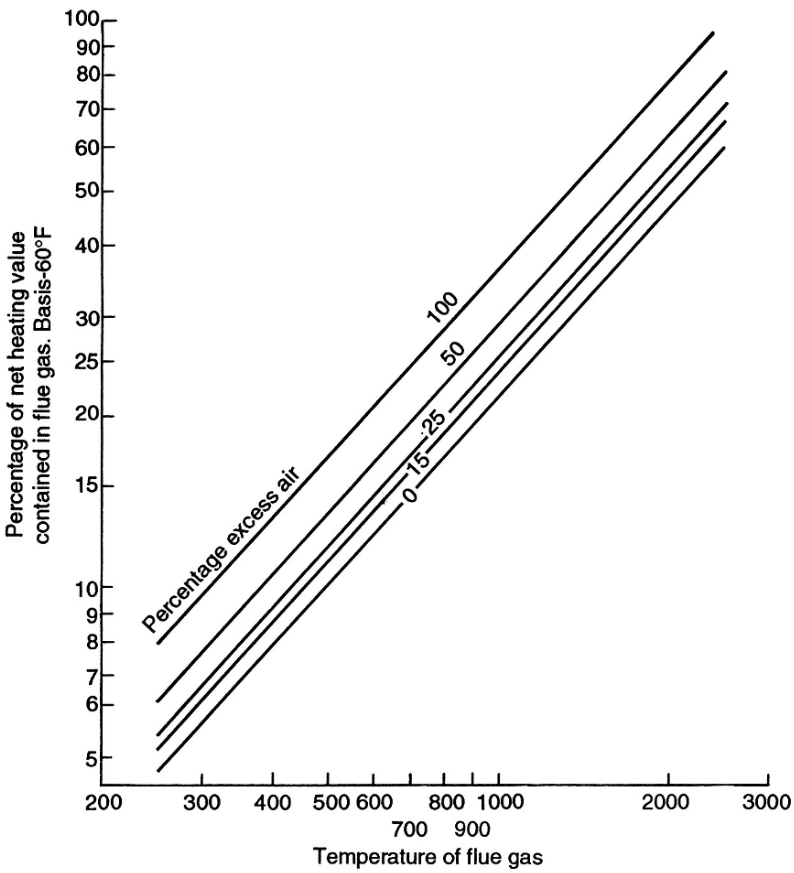


Figure 18.48. Percentage of net heating values contained in flue gas when firing fuel gas.

Step 1. Obtain details of the heater from the manufacturer's data sheet or drawings. The data required are:

- Tube area
- Layout (is it vertical and how are the burners located?)

Step 2. From plant data obtain coil inlet and outlet temperature pressures and flow. Calculate the outlet flash (i.e., vapor or liquid or a mixture of vapor and liquid) condition, then calculate its enthalpy. Do the same for the inlet flow. Usually this will be a single phase either liquid or vapor.

Step 3. Again from plant data obtain the quantity of fuel fired and its properties (API Gravity in particular).

Step 4. The difference between the enthalpies calculated in Step 2 is the enthalpy absorbed by the feed in Btu per hour.

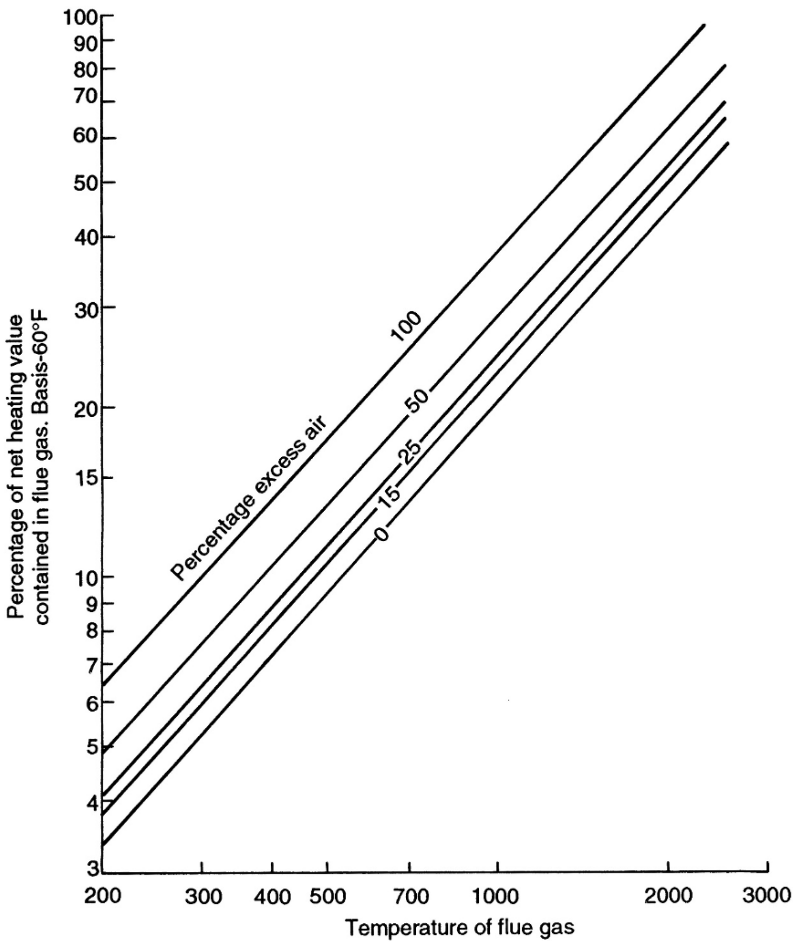


Figure 18.49. Percentage of net heating values contained in flue gas when firing fuel oil.

Step 5. Divide this absorbed enthalpy by the tube area to give the heat flux in Btu/hr/ft^2. Heat flux's are generally as follows:

Horizontal, fired one side	8,000 to 12,000 Btu/hr/ft^2
Vertical, fired from bottom on one side	9,000 to 12,000 Btu/hr/ft^2
Vertical, single row, fired on both sides	13,000 to 18,000 Btu/hr/ft^2

If the heat flux falls outside this range given above, there could be excessive fouling. Check the pressure drop—if this is far above manufacturer's calculated value then fouling is certainly present.

Step 6. Check the thermal efficiency of the heater by giving the fuel fired a heating value. This is provided by figures in the appendix. Use the LHV (lower heating valve) in Btu/lb and multiply it by the lbs/hr of the fuel.

Step 7. Divide the heat absorbed by the heat released calculated in Step 7 to give the thermal efficiency. For most heaters this should be between 70% and 80%. If it falls below this range note should be taken of burner operation and the amount of excess air being used.

Burners

The purpose of a burner is to mix fuel and air to ensure complete combustion. There are about 12 basic burner designs. These are:

Direction	—	vertical up fired vertical down fired horizontally fired
Capacity	—	high low
Fuel type	—	gas oil combination
Flame shape	—	normal slant thin, fan-shaped flat adaptable pattern
Hydrogen content	—	high
Excess air	—	normal low
Atomization	—	steam mechanical air assisted mechanical
Boiler types		
Low NO_x		
High intensity		

Various combinations of the above types are available.

Gas burners

The two most common types of gas burners are the "pre-mix" and the "raw gas" burners.

Pre-mix burners are preferred because they have better “linearity”, i.e., excess air remains more nearly constant at turndown. With this type, most of the air is drawn in through an adjustable “air register” and mixes with the fuel in the furnace firebox. This is called secondary air. A small part of the air is drawn in through the “primary air register” and mixed with the fuel in a tube before it flows into the furnace firebox. A turndown of 10:1 can be achieved with 25 psig hydrocarbon fuels. A more normal turndown is 3 : 1.

Oil burners

An oil burner “gun” consists of an inner tube through which the oil flows and an outer tube for the atomizing agent, usually steam. The oil sprays through an orifice into a mixing chamber. Steam also flows through orifices into the mixing chamber. An oil-steam emulsion is formed in the mixing chamber and then flows through orifices in the burner tip and then out into the furnace firebox. The tip, mixing chamber and inner and outer tubes can be disassembled for cleaning.

Oil pressure is normally about 140–150 psig at the burner, but can be lower or higher. Lower pressure requires larger burner tips, the pressure of the available atomizing steam may determine the oil pressure.

Atomizing steam should be at least 100 psig at the burner valve and at least 20–30 psi above the oil pressure. Atomizing steam consumption will be about 0.15–0.25 lbs steam/lb oil, but the steam lines should be sized for 0.5.

Combination burners

This type of burner will burn either gas or oil. It is better if they are not operated to burn both fuels at the same time because the chemistry of gas combustion is different from that of oil combustion. Gases burn by progressive oxidation and oils by cracking. If gas and oil are burned simultaneously in the same burner, the flame volume will be twice that of either fuel alone.

Pilots

Pilots are usually required on oil fired heaters. Pilots are fired with fuel gas.

Pilots are not required when heaters are gas fired only, but minimum flow bypasses around the fuel gas control valves are used to prevent the automatic controls from extinguishing burner flames.

Excess air and burner operation

The excess air normally used in process fired heaters is about 15–25% for gas burners and about 30% for oil burners. These excess air rates permit a wide variation in heater

firing rates which can be effectively controlled by automatic controls without fear of 'Starving' the heater of combustion air. There has been considerable work lately to reduce this excess air considerably mostly to minimize air pollution. This practice has not been used in process heaters to date. It has however been adopted in the operation of large power station type heaters with some success.

Normally companies specify that burners be sized to permit operation at up to 125% of design heat release with a turndown ratio of 3 : 1. This gives a minimum controllable rate of 40% of design without having to shut down burners.

Burner control

Burner controls become very important from safety and operation considerations. Most systems include an instrumentation system with interlocks that prohibit:

- Continuing firing when the process flow in the heater coil fails
- The flow of fuel into the firebox on flame failure

Under normal operating conditions the amount of fuel that is burnt is controlled by flow controllers operated on the coil outlet temperature. With combination burners the failure of one type of fuel automatically introduces the second type. Such a switch over can also be effected manually. This aspect is usually activated on pressure control of the respective fuel system. That is, on low pressure being sensed on the fuel being fired automatically switches to the second fuel.

Most companies operate their own specific controls for the heater firing system.

Heater noise

All heaters are noisy and this noise is the result of several mechanisms. Among these are the operation of the burners. Gas burners at critical flow of fuel emit a noise. This can be minimized by designing for low pressure drop in the system. Intake of primary and secondary air is another source of noise. Forced draft burners are generally quieter than natural draft if the air ducting is properly sized and insulated. The design of the fan can also reduce noise in this mechanism. Low tip speed fan favors low noise levels.

Refractories, stacks, and stack emissions

Refractories are used on the inside walls of the heater fire box, floor and through the convection side of the heater. The purpose of this refractory lining is to conserve the heat by limiting its loss to atmosphere by convection. It is also necessary for personal safety of those working on or about the heater who may accidentally touch the heater walls.

Good insulation has the following qualities:

- It has good high temperature strength
- It is resistant to abrasion, spalling, chemical reaction, and slagging
- It has good insulating properties

Among the most common refractories that meet some if not all of the above criteria are silica refractories, high alumina and fire clay brick. These have high resistance to spalling and to thermal shock. Their insulating qualities are also good.

Silica refractories tend to form slag with metal oxide dust and ashes. Compounds of sodium and potassium attack most refractories while refractories containing magnesium react with acids and acid gasses. Carbon monoxide and other reducing chemicals that may be present in the firebox reduce the life of refractories, particularly fire clay brick and silica.

Dense refractories with low porosity are the strongest but have the poorest insulation qualities. Castable refractories containing a mixture of cement and refractory aggregate are the cheapest and the easiest to install. They are not very rugged however. Normally in process heaters conditions are such that the use of a light insulating refractory will satisfy all that is required from a refractory lining.

The ASTM standard part 13 gives more detail on refractories. This standard provides the classification of refractories and describes their characteristics and composition. It also offers a procedure for calculating the heat loss through the insulation and thus its thickness.

Preparing refractories for operation

All new refractories need to be 'cured' after installation and before use. Refractories contain moisture, some due to the installation procedure and some in the form of water of crystallization. Curing the refractory means removing this moisture by applying a slow heating mechanism. New heater manufacturers will usually provide details of the curing procedure they recommend. The following procedure may however be used as a guide to refractory curing when manufacturers' procedures are not available:

1.0 Rise temperature at 50°F per hour to a temperature of 400°F and hold for 8 hr at that temperature.
2.0 Then rise the temperature again at 50°F per hour from 400°F to 1,000°F and hold for another 8 hr.
3.0 If necessary continue heating at 100°F per hr to the operating temperature if higher than 1,000°F. Hold at the operating temperature for a further 8 hrs.
4.0 Cool at 100°F per hr to about 500°F and hold ready for operation.

5.0 On start up the heater can be heated up to its operating temperature at a rate of 100°F per hr.

6.0 During the curing of the refractory it will be necessary to pass some fluid through the heater coils to protect them from overheating. Steam or air may be circulated through the coils for this purpose. In certain cases such as in the catalytic reforming of petroleum stock it may be necessary to circulate the nitrogen or the hydrogen that was used to purge and pressure test the unit. Air and steam in this process would not be desirable.

Stacks

Stacks are used to create an updraft of air from the firebox of a heater. The purpose of this is to cause a small negative pressure in the firebox and thus enable the introduction of air from the atmosphere. This negative pressure also allows for the removal of the products of combustion from the firebox. The stack therefore must have sufficient height to achieve these objectives and overcome the frictional pressure drop in the firebox and the stack itself.

The height required for a stack to achieve good draft can be estimated from the following equation:

$$D = 0.187H(\rho_a - \rho_g)$$

where

D = draft in ins of water.
H = Stack height in feet.
ρ_a = Density of atmospheric air in lbs/cuft.
ρ_g = Density of stack gasses in lbs/cuft at stack conditions.

For stack gas temperature use 100°F lower than gasses leaving the convection section. Stack gasses have a molecular weight close to that of nitrogen. For this calculation use 28 as the mole weight. Burner draft requirements range from 0.2 to 0.5 ins of water. Use 0.3 ins of water as a good design value.

Stacks must also be designed to handle and disperse stack emissions. This usually results in having to build stack heights greater than that required for obtaining draft. There are available specific computer programs relating plume height of the stack gasses above the stack outlet to the probable ground level fallout of the impurities in the gas. These programs also produce a map of the relative concentrations of these impurities at ground level. Such data is usually available from government authority offices in most countries. The predicted ground concentration and map calculated are checked against local legal requirements. The results usually form part of the government approval to build and/or operate a facility.

The stack diameter is based on an acceptable velocity of stack gasses in the stack. This is generally taken as 30 ft/sec. Some allowance must be made for frictional losses these are:

1.5 velocity heads for inlet and outlet losses.
1.5 velocity head for damper.
1.0 velocity head for each 50 ft of stack height.

Stack emissions

The obnoxious compounds in stack gas emissions arise from:

- Impurities in the fuel
- Chemical reactions resulting from the fuel combustion with air

The three major impurities in oil or gas fuels which produce undesirable emissions are:

- Sulfur
- Metals
- Nitrogen

Sulfur

All gas and oil fuels contain sulfur at some level of concentration. When these fuels are burned the sulfur reacts with air to form SO_2 and SO_3. These compounds are objectionable because they cause:

- Air pollution in the form of smog
- They contribute to the corrosion of heater tubes and stack
- SO_3 lowers the dew point of the flue gasses resulting in an objectionable visible plume at the stack exit

Figure 18.50 shows the effect of sulfur in the feed on the flue gas dew point. This dew point also of course varies with the amount of excess air and the relative partial pressure of the combustion gasses.

There is no precise way of determining the amount of SO_3 that is formed in the flue gas. Nor can it be determined with any degree of accuracy where the SO_3 is formed in the system. The total amount of the sulfur oxides is of course determined simply from the sulfur content of the feed. Because of this uncertainty it is important to restrict the minimum allowable convection side metal temperatures to 350°F when firing fuels containing sulfur. Also the minimum temperatures to the stack should be 320°F when firing fuel gas and 400°F when firing fuel oil. In the case of metal stacks the use of a non-corrosive lining must be used in the colder section of the stack if the flue gas temperature falls below those stated above.

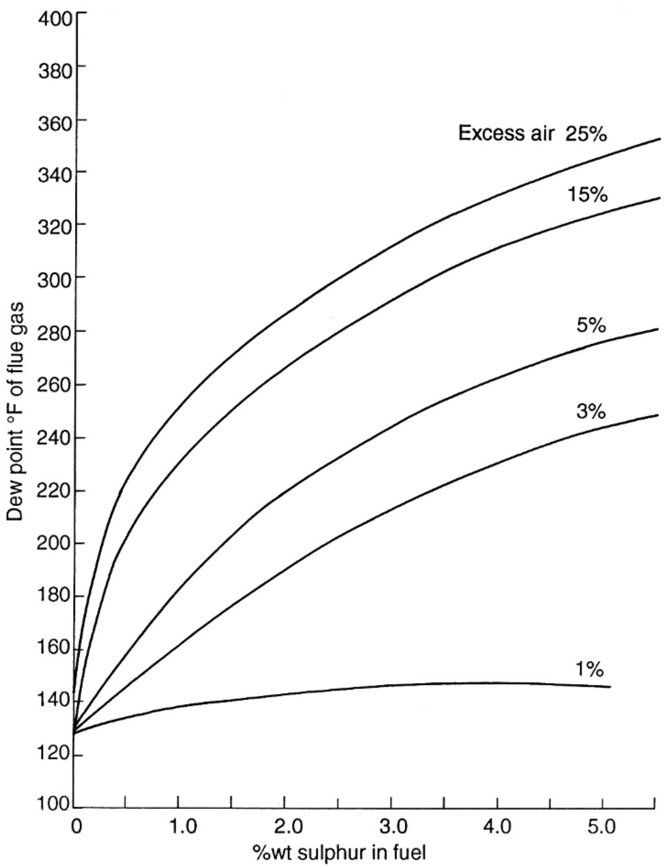


Figure 18.50. Dew point of flue gases versus sulfur content.

Metals

The most objectionable metal impurities in fuel oil are sodium and vanadium. These can cause severe corrosion of tubes and refractory lining. By high vanadium content is meant concentrations between 200 ppm and 400 ppm, above 400 ppm is considered very high and should not be used as a fuel. Vanadium in the presence of sodium and oxygen readily forms a corrosive compound:

$$Na_2O \cdot 6V_2O_5$$

This compound attacks iron or steel to form ferric oxides according to the equation:

$$Na_2O \cdot 6V_2O_5 + Fe \longrightarrow Na_2O\ V_2O_4 \cdot 5V_2O_5 + FeO$$

This vanadium product is oxidized back to the original corrosive compound according to the reaction:

$$Na_2OV_2O_4 \cdot 5V_2O_5 + O_x \longrightarrow Na_2O \cdot 6V_2O_5$$

These reactions occur at temperatures between 1,070°F and 1,220°F with the vanadium compound being continually regenerated to its corrosive state.

Sodium and sulfur also combine to form undesirable corrosive compounds without vanadium being present. This reaction forms sodium iron tri-sulfate $Na_3Fe(SO_4)_3$ at temperatures of around 1,160°F. The critical temperature for both vanadium and sodium corrosion is around 1,100°F. Vanadium is not a major problem at this temperature but becomes so at temperatures above this level.

Nitrogen

Some nitrogen oxides will be formed in combustion gasses even if there are no nitrogen compounds in the fuel. The presence of the nitrogen compounds in the fuel significantly increases the nitrogen oxide content of the flue gas. There are six oxides of nitrogen present in flue gasses but only two are present in any appreciable amount. These are:

NO Nitrogen Oxide.
NO_2 Nitrogen Dioxide.

Nitrogen oxide is a poisonous gas which readily oxidizes to nitrogen dioxide on entering the atmosphere. Nitrogen dioxide is a yellowish gas which readily combines with the moisture in the air to form nitric acid. In the presence of sunlight and oxygen nitrogen dioxide also contributes to the formation of other air pollutants collectively labeled NO_x. The production of NO_x can be reduced by limiting the amount of excess air, by washing the flue gasses with aqueous ammonia, or by catalytically reducing NO_x in the presence of ammonia.

Specifying a fired heater

Some basic data concerning a fired heater must be made known before the equipment can be designed for fabrication or even costed. These data are provided by a specification sheet or sheets. In the case of a fired heater as in the case for compressors this specification can run into several sheets or forms. Such sheets will define the unit in terms of:

- Process Requirement
- Mechanical Detail
- Civil engineering Requirement
- Operational Requirement
- Environmental Requirement

FIRED HEATER.

Design Duty 40886000 Btu\hr Service Hydrocarbons

No Heaters 1 Unit Vac Unit Preheater

Item No H 201 Type Horizontal

DESIGN DATA :-	Radiant sec	Conv sec
Service	Red Crude	Steam s.heat
Heat Absorption mm Btu\hr	38.501	2.385
Fluid	Hydrocarbon	Sat Steam
Flow Rate lbs / hr	197085	30040
Allowable Pressure drop PSI	300	30
Allowable average Flux Btu / hr. sqft	15000	____
Maximum Inside Film Temp °F	800	_____
Fouling Factor °F. sqft. hr / Btu	0.004	0.001
Residence Time sec	N/A	N/A
Inlet Conditions		
Temperature °F	554	368
Pressure PSIG	270	155
Liquid Flow lbs / hr	197085	nil
Vapor Flow lbs/ hr	nil	30040
Liquid Density lbs / cuft	48.4	___
Vapour Density lbs\cuft	______	Steam
Visc Liq\Vap Cps	2.31 / ___	__ / ___
Specific Heats liq\vap Btu\lb	0.65 /___	/
Thermal Cond liq\vap Btu\hr.sqft.°F\Ft	0.0671 /___	/

Figure 18.51. Specification sheet for fired heaters.

This item will deal only with the "Process Requirement" and the duty of the heater. A typical process originated specification sheet is shown here as Figure 18.51. The data provided on this sheet are the minimum that will be necessary for a heater manufacturer or a heater specialist to begin to size the heater or price out the item. Usually this data would also be supported with a vaporization curve of the feed if there is a change of phase taking place in the heater coils. The process engineer developing this specification would also provide details of services required in addition to the main duty of the heater. For example if it is intended that the convection side of

FIRED HEATER. (Cont)

DESIGN DATA (Cont)	Rad Sec	Conv Sec
Outlet Conditions.		
Temperature °F	750	500
Pressure (Psia) PSIG	(0.68)	125
Liquid Flow lbs\hr	69641	nil
Vapour Flow lbs\hr	127444	30040
Liquid Density lbs\cuft	47.3	_____
Vapour Density lbs\cuft	0.023	0.246
Visc of Liq\Vap Cps	1.4 /0.002	__ /___
Specific Heat Liq\Vap Btu\lb	0.64 /0.69	___ /___
Thermal Cond Liq\Vap Btu\hr.sqft.°F\ Ft	0.062 /0.023	/
FUEL DATA		
Type (Gas or Oil)	OIL	GAS
LHV Gas Btu\cuft Oil Btu\lb	17560	2320
HHV Gas Btu\cuft Oil Btu\lb	18580	2520
Pressure at Burner PSIG	82	25
Temp at Burner °F	176	68
Mol Wt of Gas		44
Visc of Oil @ Burner Cps	23.3	
Atomising Steam Temp°F Press PSIG	500 125	
Composition of Gas Mol %		
H2		0.2
C1		12.0
C2		28.2
C3		31.3

Figure 18.51. (Cont.)

the heater is to be used for steam generation, preheating, or steam superheating, the system envisaged must be properly described by additional diagrams and data.

The example specification sheet Figure 18.51 is completed for a crude oil vacuum distillation unit heater with a steam super heater coil located in the convection side of the heater. The following paragraphs describe the content on a line by line basis:

FIRED HEATER (Cont).

FUEL DATA (Cont)		
Composition of Gas (Cont)		
C4's		
C5's		
C6+		
Properties of Fuel oil		
° API	15.2	
Visc @ 100°F cs	175	
Visc @ 210°F cs	12.5	
Flash Pt °F	200	
Vanadium ppm	12	
Sodium ppm	32	
Sulphur %wt	2.4	
Ash %wt	< 1.0	

REMARKS

1.0 Flash curve of the reduced crude feed versus % volume distilled is attached for atmos pressure and at 35mm Hg Abs.
2.0 Gravity and mole wt curves for the reduced crude versus mid volume % are attached.
3.0 Soot blowers are to be considered in this package. Steam is available at 600 psig and 750°F.
4.0 Studded tubes to be considered for the convection side.

Figure 18.51. (Cont.)

Line 1. *Design duty* refers to the total duty required of the unit. In the case of the example given here it includes the duty required to heat and partially vaporize the oil in the radiant section and the duty required to superheat the steam. The data sheet will be split into two sections to reflect each of these duties.

Service describes the main purpose for which the heater will be used. In this case it will be used to heat and vaporize hydrocarbons.

Line 2. *No of heaters.* This is self explanatory. In this case there is only one heater required. Should there have been more than one identical unit this would be reflected here.

Unit. This is the title given to this unit of equipment as it appears on an equipment list. It will correspond to the item number also given in the equipment list. In the case of the example it is "Vac unit pre-heater."

Line 3. *Item No.* This is the reference number given to the item in the equipment list. This reference number and unit title identifies the equipment on all drawings where it appears, and all documents used in its purchase, costing, maintenance, etc.

Type. The type of heater (if decided on) is given here. In the case of the example heater a cabin (Horizontal Tubes) type has been selected for vacuum services considerations.

Line 4. This is the first line of the specification sheet proper. It commences with the service of the heater or section of the heater. In the case of this example only two columns have been provided. These are designated for the 'Radiant' section and 'Convection' section. On most preprinted specification forms there would be at least 4 columns.

Line 5. *Heat absorption.* This line divides the duty of the heater into that required from the radiant coils and that required from the convection side. In this example the oil is routed through the radiant section only while saturated steam from a waste heat boiler is superheated in the convection coils. Both are measured in million Btu/hr.

Partial vaporization of the oil occurs in the radiant coil. Therefore a flash curve or a phase diagram of the oil must accompany this specification sheet. In the example the duty to the oil is calculated from data developed in the material balance and heat balance of the process. Thus:

Temperature of the feed into the heater is 554°F and that of the coil outlet is 750°F. From the flash curve at the outlet pressure of 35 mm Hg abs (0.68 psia) and the material balance the weight per hr of vapor is calculated to be 127,444 lbs/hr and the liquid portion is 69,641 lbs/hr. The heat absorbed in the radiant section is:

$$
\begin{aligned}
\text{Heat in with feed} &= 197{,}085 \text{ lbs/hr} \times 268 \text{ Btu/lb (all liquid)} \\
&= 52.819 \text{ mm Btu/hr.} \\
\text{Heat out in feed:} & \\
\text{Liquid portion} &= 69{,}641 \text{ lbs/hr} \times 378 \text{ Btu/lb} \\
&= 26.324 \text{ mm Btu/hr.} \\
\text{Vapor portion} &= 127{,}444 \text{ lbs/hr} \times 510 \text{ Btu/llb} \\
&= 64.996 \text{ mm Btu/hr.} \\
\text{Total heat out} &= 91.320 \text{ mm Btu/hr.} \\
\text{Duty of radiant sect} &= 91.320 - 26.324 \\
&= 38.501 \text{ mm Btu/hr.}
\end{aligned}
$$

For the convection section the duty is calculated as follows:

Weight of saturated steam at 155 psig is 30,040 lbs/hr

$$
\begin{aligned}
\text{Temperature of 155 psig steam} &= 368^\circ\text{F} \\
\text{From steam tables steam enthalpy} &= 1{,}195.9 \text{ Btu/lb} \\
\text{Temperature of steam out} &= 500^\circ\text{F} \\
\text{Pressure of steam out} &= 125 \text{ psig.} \\
\text{From steam tables steam enthalpy} &= 1{,}275.3 \text{ Btu/lb.} \\
\text{Duty of convection side} &= 30{,}040\,(1{,}275.3 - 1{,}195.9) \\
&= 2.385 \text{ mm Btu/hr.}
\end{aligned}
$$

Line 6. *Fluid.* This refers to the material flowing in the coil. In the case of this example it will be hydrocarbons in the radiant coil and steam in the convection coil.

Line 7. *Flow rate.* This is the total flow rate in lbs per hour entering the respective section of the heater. Thus for the radiant side the figure will be 197,085 lbs/hr, and for the convection side it will be 30,040 lbs/hr.

Line 8. *Allowable pressure drop.* The process engineer enters the required pressure drop calculated from the hydraulic analysis of the system. This pressure drop is measured from the heater side of the inlet manifold down stream of the balancing control valves and the coil outlet down stream of the outlet manifold.

Line 9. *Allowable average flux.* This is usually a standard set by the company for its various heaters. In the example here this value would be between 13,500 and 18,000 Btu/hr sqft. (a horizontal heater fired on both sides). It is specified as 15,000 Btu/hr sqft for this example and refers only to the radiant section.

Line 10. *Maximum inside film temperature.* It is important to notify the heater manufacturer of any temperature constraint that is required by the process. In the case of this example temperatures of the oil above 800°F may lead to the oil cracking. Such a situation could adversely affect the performance of the downstream fractionation equipment and therefore high temperatures in excess of 800°F must be avoided. There is no constraint on the convection coil.

Line 11. *Fouling factor.* The fouling factors used in heat exchanger rating can be used here also. In this example therefore the radiant side would have a fouling factor of about 0.004°F sqft hr/Btu for the oil and 0.001 for the steam.

Line 12. *Residence time.* This becomes important when a chemical reaction of any kind takes place in the heater tubes. In the case of this example this item does not apply. If the example were a thermal cracking heater or a visbreaker the appropriate kinetic equations and calculations would be attached to the specification sheet to support this item.

Line 13–30. These are self explanatory. The only comment here is that the data are quoted at the inlet or outlet conditions of temperature and pressure.

Line 31. *LHV.* The section of the specification sheet that follows deals with the characteristics of the fuel that will be used in the heater. This section is divided into oil and gas which are the usual fuels used in modern day processes. This item requires the "Lower heating value" of the fuel. This can be read off charts such as Figures A9.2 and A9.3.

Line 32. *HHV.* This is the other heating value data required by the heater designer. This "higher heating value" data can also be read off charts such as Figures A9.2 and A9.3.

Line 33. *Pressure at burner.* This normally refers to the oil fuel and is measured at the heater fuel oil manifold.

Line 34. *Temperature at the burner.* This item too is self explanatory and these measurements are also taken at the respective manifolds.

Line 35. *Mol wt of gas.* This refers to the gas stream normally expected to be used. Obviously in practice this will vary with the process operation from day to day.

Line 36. *Viscosity of the oil at the burner.* This viscosity is quoted at the burner temperature and may be arrived at from the two viscosity figures given later in the specification sheet. This is important for the best design or selection of the burner itself.

Line 37. *Atomizing steam.* This item calls for the temperature and pressure of the steam that will be used for atomizing the oil fuel. This is also required for the best design or selection of the oil burner.

Line 38. *Composition of gas.* This section requires the composition of the gas fuel in terms of mol percent. This is the normal expected fuel gas that will be used in the heater. If there is likely to be a wide variation in the quality of the fuel gas that will be used this should be noted here as a range of two or even three compositions. This situation is particularly common in petroleum refining.

Line 46. *Properties of fuel oil.* This final section of the specification sheet requires details of the fuel oil that will be used. These details are:

Gravity of the oil @ 60°F (in °API).
Viscosity of the oil @ 100°F and at 210°F
Flash point in °F.

From the two viscosities the 'Refutus' Graph can be used to determine the viscosity at any other temperature. This graph can be found in most data books that carry viscosity data. Note the viscosity requested in this section is in centistokes (kinematic viscosity). This is because most suppliers quote in centistokes. To convert to centipoise multiply by the specific gravity (grams/cm^3).

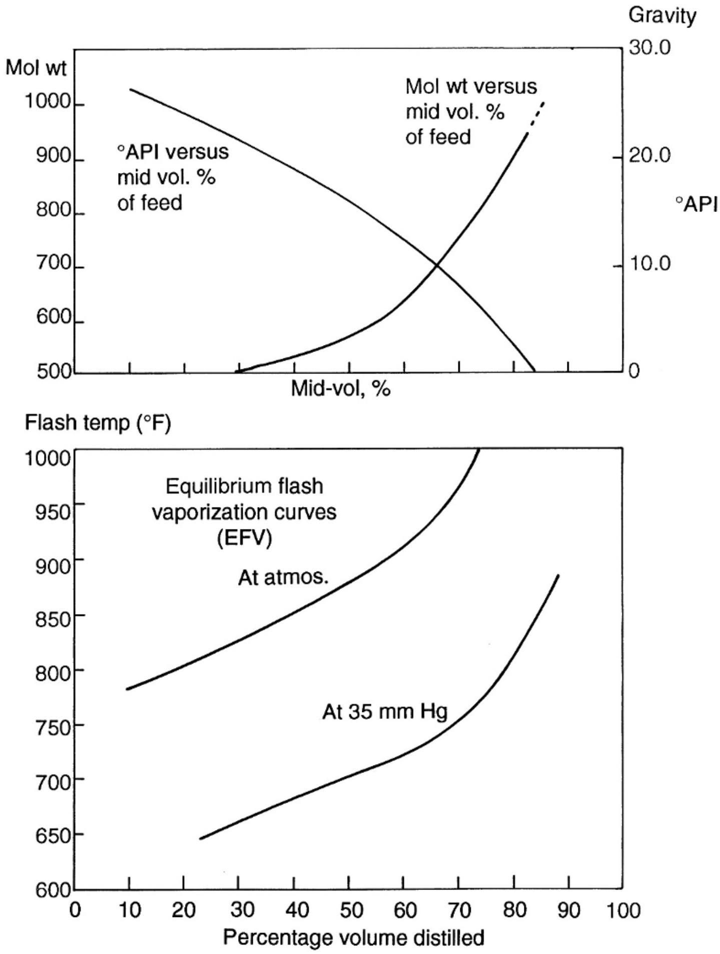


Figure 18.52. Specification sheet attachment.

The flash point required is that determined by the Pensky Marten method and is a measure of the oil's flammability.

This completes the explanation of the main body of the specification sheet. It represents the minimum data required to commence the sizing of the item. The last part of the specification sheet is given to "REMARKS". In this section the engineer should provide all the other data that may influence the design of the heater, e.g. Figure 18.52.

APPENDICES

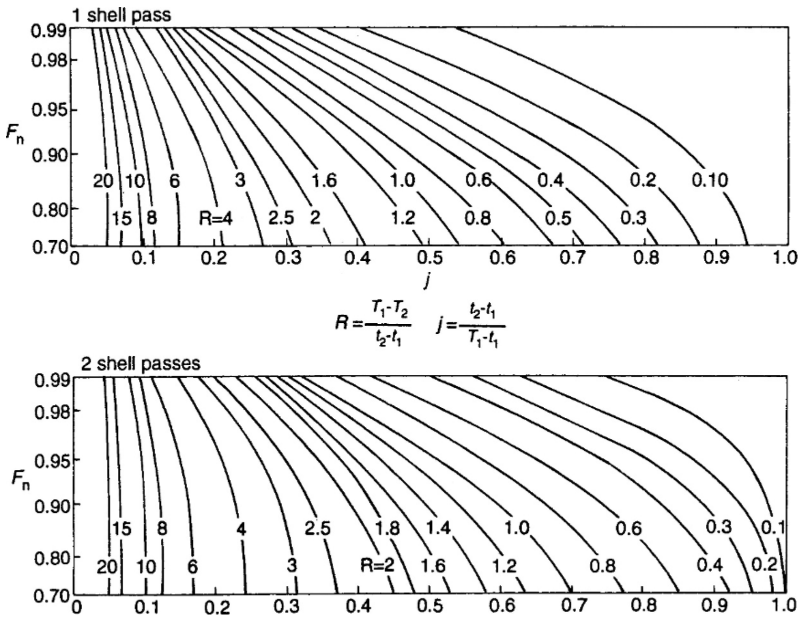
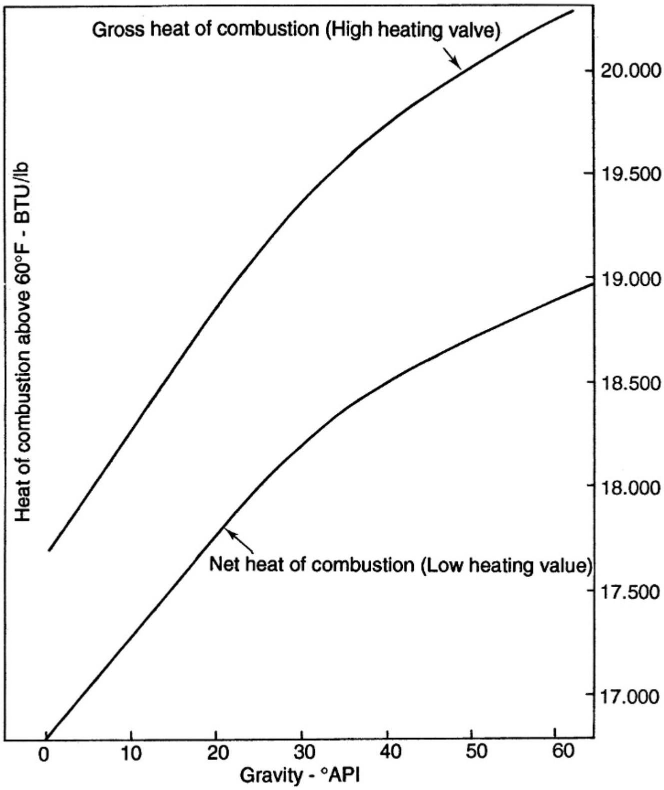


Figure 18.A.1. LMTD correction factors.



Impurities in average fuels			
°API	%S	%inerts	Total impurity
Residual fuel oil and crudes			
0	2.95	1.15	4.10
5	2.35	1.00	3.35
10	1.80	0.95	2.75
15	1.35	0.65	2.20
20	1.00	0.75	1.75
Crude oils			
25	0.70	0.70	1.40
30	0.40	0.65	1.10
35	0.30	0.60	0.90

Figure 18.A.2. Heat of combustion of fuel oils.

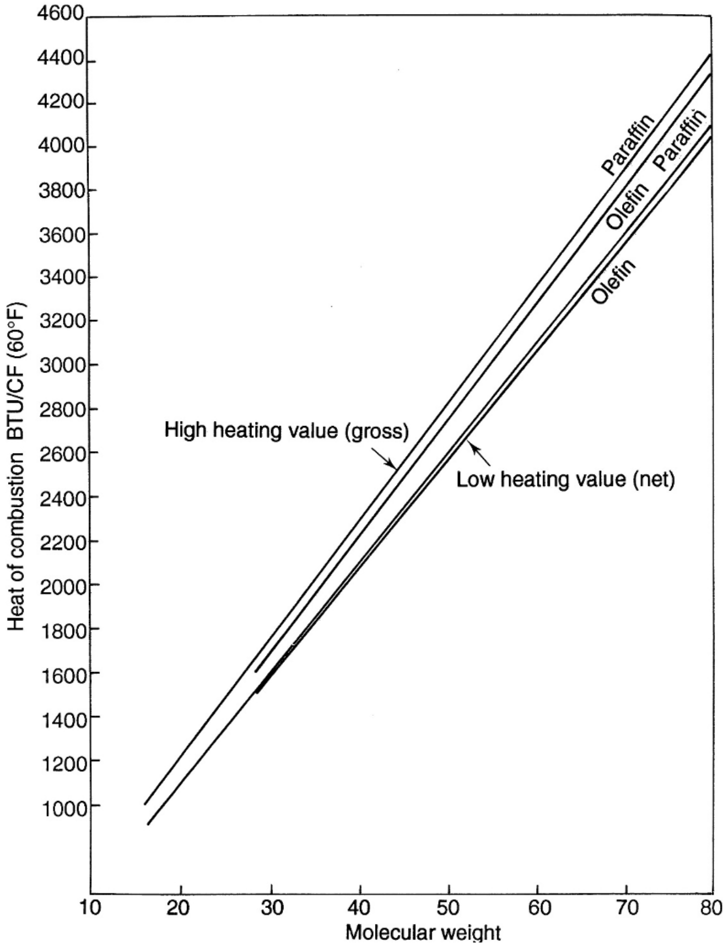


Figure 18.A.3. Heat of combustion of fuel gasses.

Table 18.A.1. Values for coefficient C.

Values of coefficient $C = 520\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}$; $k = \frac{C_p}{C_v}$

k	C	k	C	k	C	k	C	k	C	k	C
0.41	219.28	0.71	276.09	1.01	316.56*	1.31	347.91	1.61	373.32	1.91	394.56
0.42	221.59	0.72	277.64	1.02	317.74	1.32	348.84	1.62	374.09	1.92	395.21
0.43	223.86	0.73	279.18	1.03	318.90	1.33	349.77	1.63	374.85	1.93	395.86
0.44	226.10	0.74	280.70	1.04	320.05	1.34	350.68	1.64	375.61	1.94	396.50
0.45	228.30	0.75	282.20	1.05	321.19	1.35	351.60	1.65	376.37	1.95	397.14
0.46	230.47	0.76	283.69	1.06	322.32	1.36	352.50	1.66	377.12	1.96	397.78
0.47	232.61	0.77	285.16	1.07	323.44	1.37	353.40	1.67	377.86	1.97	398.41
0.48	234.71	0.78	286.62	1.08	324.55	1.38	354.29	1.68	378.61	1.98	399.05
0.49	236.78	0.79	288.07	1.09	325.65	1.39	355.18	1.69	379.34	1.99	399.67
0.50	238.83	0.80	289.49	1.10	326.75	1.40	356.06	1.70	380.08	2.00	400.30
0.51	240.84	0.81	290.91	1.11	327.83	1.41	356.94	1.71	380.80	2.01	400.92
0.52	242.82	0.82	292.31	1.12	328.91	1.42	357.81	1.72	381.53	2.02	401.53
0.53	244.78	0.83	293.70	1.13	329.98	1.43	358.67	1.73	382.25	2.03	402.15
0.54	246.72	0.84	295.07	1.14	331.04	1.44	359.53	1.74	382.97	2.04	402.76
0.55	248.62	0.85	296.43	1.15	332.09	1.45	360.38	1.75	383.68	2.05	403.37
0.56	250.50	0.86	297.78	1.16	333.14	1.46	361.23	1.76	384.39	2.06	403.97
0.57	252.36	0.87	299.11	1.17	334.17	1.47	362.07	1.77	385.09	2.07	404.58
0.58	254.19	0.88	300.43	1.18	335.20	1.48	362.91	1.78	385.79	2.08	405.18
0.59	256.00	0.89	301.74	1.19	336.22	1.49	363.74	1.79	386.49	2.09	405.77
0.60	257.79	0.90	303.04	1.20	337.24	1.50	364.56	1.80	387.18	2.10	406.37
0.61	259.55	0.91	304.33	1.21	338.24	1.51	365.39	1.81	387.87	2.11	406.96
0.62	261.29	0.92	305.60	1.22	339.24	1.52	366.20	1.82	388.56	2.12	407.55
0.63	263.01	0.93	306.86	1.23	340.23	1.53	367.01	1.83	389.24	2.13	408.13
0.64	264.72	0.94	308.11	1.24	341.22	1.54	367.82	1.84	389.92	2.14	408.71
0.65	266.40	0.95	309.35	1.25	342.19	1.55	368.62	1.85	390.59	2.15	409.29
0.66	268.06	0.96	310.58	1.26	343.16	1.56	369.41	1.86	391.26	2.16	409.87
0.67	269.70	0.97	311.80	1.27	344.13	1.57	370.21	1.87	391.93	2.17	410.44
0.68	271.33	0.98	313.01	1.28	345.08	1.58	370.99	1.88	392.59	2.18	411.01
0.69	272.93	0.99	314.19*	1.29	346.03	1.59	371.77	1.89	393.25	2.19	411.58
0.70	274.52	1.00	315.38*	1.30	346.98	1.60	372.55	1.90	393.91	2.20	412.15

Values of C for gases

	Mol wt	$k = C_p/C_v$	C	$C/356$		Mol wt	$k = C_p/C_v$	C	$C/356$
Acetylene	26	1.28	345	0.969	Hydrochloric acid	36.5	1.40	356	1.000
Air	29	1.40	356	1.000	Hydrogen	2	1.40	356	1.000
Ammonia	17	1.33	351	0.986	Hydrogen sulphide	34	1.32	348	0.978
Argon	40	1.66	377	1.059	Iso-butane	58	1.11	328	0.921
Benzene	78	1.10	327	0.919	Methane	16	1.30	346	0.972
Carbon disulphide	76	1.21	338	0.949	Methyl alcohol	32	1.20	337	0.947
Carbon dioxide	44	1.28	345	0.969	Methyl chloride	50.5	1.20	337	0.947
Carbon monoxide	28	1.40	356	1.000	N-butane	58	1.11	328	0.921
Chlorine	71	1.36	352	0.989	Natural gas	19	1.27	345	0.969
Cyclohexane	84	1.08	324	0.910	Nitrogen	28	1.40	356	1.000
Ethane	30	1.22	339	0.952	Oxygen	32	1.40	356	1.000
Ethylene	28	1.20	337	0.947	Pentane	72	1.09	325	0.913
Helium	4	1.66	377	1.059	Propane	44	1.14	331	0.930
Hexane	86	1.08	324	0.910	Sulphur dioxide	64	1.26	342	0.961

* Interpolated values, since C becomes indeterminate as k approaches 1.00. (Reproduced by permission of Gas Processors Suppliers Association)

Table 18.A.2. Some common heat transfer coefficients Uo.

Fluid being cooled	Fluid being heated	$\frac{U_0}{(\mathrm{BTU/h.f^2.^\circ F})}$
Exchangers		
C_4s and lighter	Water	75–110
C_4s and lighter	LPG	75
Naphtha	Naphtha	75
Naphtha	Water	80–100
BPT 450 (kero)	Hy oil (crude)	70–75
Gas oils	Crude	40–50
Gas oils	Water	40–70
Light fuel oil	Crude	20–30
Waxy distillates	Hy oils	30–40
Slurries	Waxy distillate	40
MEA or DEA	Water	140
MEA or DEA	MEA or DEA	120–130
Water	Water	180–200
Air	Water	20–30
Lt HC vapour	H_2-rich stream	35–40
Lt HC vapour	Naphtha	38
Condensers		
Full-range naphtha	Water	70–80
Amine stripper O/heads	Water	100
C_4s and lighter	Water	90
Reformer effluent (Lt HC)	Water	65

Table 18.A.3. Standard exchanger tube sheet data.

d_0 = o.d. of tubing (in.)	BWG gauge	l = thickness (ft)	d_i =i.d. of tubing (in.)	Internal area (in.^{2})	External surface per foot length (ft^2)
$\frac{3}{4}$	10	0.0112	0.482	0.1822	0.1963
$\frac{3}{4}$	12	0.00908	0.532	0.223	0.1963
$\frac{3}{4}$	14	0.00691	0.584	0.268	0.1963
$\frac{3}{4}$	16	0.00542	0.620	0.302	0.1963
$\frac{3}{4}$	18	0.00408	0.652	0.334	0.1963
1	8	0.0137	0.670	0.355	0.2618
1	10	0.0112	0.732	0.421	0.2618
1	12	0.00908	0.782	0.479	0.2618
1	14	0.00691	0.834	0.546	0.2618
1	16	0.00542	0.870	0.594	0.2618
1	18	0.00408	0.902	0.639	0.2618
$1\frac{1}{2}$	10	0.0112	1.232	1.192	0.3927
$1\frac{1}{2}$	12	0.00908	1.282	1.291	0.3927
$1\frac{1}{2}$	14	0.00691	1.334	1.397	0.3927
$1\frac{1}{2}$	16	0.00542	1.37	1.474	0.3927